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A High Temperature Heat Pump Dehumidification System for use in a Cloth Drying Oven

Sunwell Engineering Company Limited

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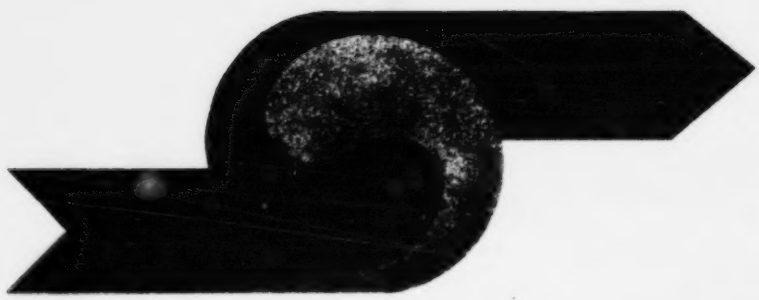

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Sunwell Engineering Company Limited

A HIGH TEMPERATURE HEAT PUMP DEHUMIDIFICATION SYSTEM FOR USE IN A CLOTH DRYING OVEN

Prepared for:
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1.0 EXECUTIVE SUMMARY

The object of the present study was to design a high temperature heat pump dehumidification system which could be added to the existing Canada Sandpaper facility in Plattsville, Ontario with the intention to reduce energy costs. It involved the physical evaluation of the existing steam-heated cloth dryer, the theoretical aspects of the heat pump dehumidification system, the design and construction of the proposed system. A theoretical analysis has been performed on the processes of evaporation of moisture from the wet cloth and dehumidification of the moisture from the oven air.

The system presently in use at Canada Sandpaper requires approximately 8,020,000 Btu/hr. solely to preheat the make-up air to the oven. Both the latent and sensible portions of heat in the oven are being lost through the exhaust. Furthermore, pollutants resulting from evaporation of moisture from raw cloth are being exhausted into the environment.

The proposed heat pump dehumidification system will allow the oven to operate at a lower temperature and it will completely eliminate the need for exhaust. The only energy input to the system will be that required by the compressors, the blowers, fans for the condenser and evaporator.

According to our estimates, the saving in energy costs based on an operating time of 4000 hours per year is approximately $82,250. Although the capital cost of the heat pump dehumidification system is estimated at $250,000 yet the payback period is estimated at approximately 3 years.

It has been estimated that of the energy consumed by the British Industry, other than the Iron and Steel sector, approximately 6% is used simply to vaporize water.

From this implication we can assume with confidence that our industry in Canada is facing the same problem. The heat pump dehumidification system is naturally the best solution to reduce that 6% waste of energy.

RÉSUMÉ POUR LA DIRECTION

L'objet de la présente étude était de concevoir un système de déshumidification à pompe à chaleur haute température destiné aux installations de la Canada Sandpaper de Plattsville (Ontario), dans le but de réduire les coûts d'énergie de ces installations. L'étude a comporté plusieurs volets: évaluation du séchoir existant chauffé à la vapeur, aspects théoriques du système projeté, puis conception et construction de ce système. On a aussi fait une analyse théorique des procédés d'évaporation de l'eau de la toile humide et de déshumidification de l'air du four.

Dans les installations qu'exploite actuellement la Canada Sandpaper, il faut environ 8 020 000 BTU/h uniquement pour préchauffer l'air d'appoint du four. La chaleur latente et la chaleur sensible de ce dernier sont perdues dans l'évacuation. Enfin, les polluants produits par l'évaporation de l'eau de la toile sont rejetés dans l'environnement.

Le système projeté permettra de faire fonctionner le four à une température moindre et en supprimera l'évacuation. Le seul apport d'énergie sera celui qui servira à alimenter les compresseurs, les soufflantes et les ventilateurs du condenseur et de l'évaporateur.

D'après nos chiffres, les économies d'énergie seront de 82 250 $ pour un temps d'exploitation annuel de 4000 heures. On estime que la période d'amortissement sera de trois ans approximativement, malgré l'importance de l'investissement qu'on évalue à 250 000 $.

Il a par ailleurs été estimé que de toute l'énergie consommée par l'industrie britannique, exception faite de la sidérurgie, 6 % environ servent simplement à faire évaporer de l'eau.

On peut raisonnablement supposer que la situation est identique au Canada. Le système projeté est naturellement la meilleure solution pour réduire ce gaspillage d'énergie.

2.0 PHYSICAL EVALUATION OF THE EXISTING SYSTEM

2.1 Objective

The objective involves a complete physical evaluation of the existing cloth dryer at the Canada Sandpaper facility in Plattsville, Ontario.

It also includes operational tests of the existing system to determine temperature and humidity conditions within the oven environment and in the exhaust air, cloth moisture content before and after drying, and present energy consumption.

2.2 Introduction - Physical Evaluation of the Existing system

At the Canada Sandpaper facility in Plattsville, Ontario a denim type of cloth is used as the backing for one type of sandpaper. The denim cloth must be washed, stretched, and dried in order to prevent shrinkage during subsequent production operations.

This denim cloth, usually in the form of 3 feet diameter rolls, is set up to be processed in a continuous operation. After being cleaned in the cloth washer with 200 degree F. water, it is squeezed through a set of press rollers to remove the major portion of the moisture. Drying is performed in the steam-heated tenter dryer which is manufactured by Carborundum Co. in New York. This dryer measured 45'-6" in length, 12'-6" in width and 10'-4" in height, comprised of two heating zones. Zone 1 is for pre-heating and zone 2 for drying. Hot air at 250 degrees F. is blown from the evenly distributed nozzles on to the wet cloth. Nozzle jets are located in such a way that equal amounts of hot air are applied on both the top and bottom surfaces of the wet cloth. Drying is achieved by

forced convection with hot air to facilitate evaporation of moisture from wet surfaces. As the relative humidity of the air would increase in the oven with the lowering of temperature, this hot air must be exhausted at a rate of 30,000 CFM. To compensate for the loss of hot air through the exhaust, make-up air must be added to the oven.

As the oven air is continuously being diluted by mixing with the make-up air, air handling units are required to heat up this air to a temperature of 250 degrees F. Steam coils are used for this purpose. As a result, both the latent and sensible portions of the oven heat are being lost through the exhaust. Furthermore, pollutants resulting from evaporation of moisture from the raw cloth are being exhausted into the environment.

2.3 Description of Cloth Dryer Physical Evaluation of the Existing System

The basic components of this steam heated tenter dryer are as follows:

(1) Four 17,000 CFM fans driven by 15 H.P. 1800 RMP TEFC motors. Each air handling unit is powered by one of these fans.

(2) Two 10,300 CFM fans driven by 3 H.P. 18000 RPM TEFC motors. Each exhaust unit is powered by one of these fans.

(3) Four steam coils, each composed of 39 tubes of 1/2" diameter, provide the heat energy for the air-handling units.

(4) 56 Nozzle jets with outlet openings of 1/2" x 65-1/2" are distributed evenly above and below the conveyor - driven wet cloth in both heating zones.

(5) Conveyor system to carry the cloth first through the cleaning tank and then the drying oven. This operation is continuous.

(6) Mild carbon steel ductworks and shroudings to make up the two heating zones of the drying oven.

2.4 Theory - Physical Evaluation of the Existing System

The operation to wash, stretch and dry the backing cloth is designed to be continuous and automatic. After being cleaned in the cloth washer with 200 degree F. water, the cloth is wetted with a pre-determined amount of water. Passing through the set of press rollers would remove a large portion of the moisture from the wet cloth. After processing through the drying oven, this cloth is allowed to contain 10% of its initial moisture. The cloth must be fed through the drying oven at a certain speed such that 10% of the moisture would remain in the material. Adjustment of the feed speed is required for different types of backing cloth.

In the drying oven, the operating temperature of air is controlled by the temperature sensing device. In the first preheating zone, 28 nozzles are placed on both the top and bottom sides of the wet cloth. Hot air at a temperature of 250 degrees F. is blown on the top and bottom surfaces of the cloth. This hot air raises the temperature of the cloth and provides the latent heat to evaporate part of the moisture carried by the cloth, by forced convection. In the second drying zone, the rest of the mofsture is evaporated by forced

convection until only 10% of its original content remains in the material.

Energy required for pre-heating and evaporation of moisture.

Evaporation of moisture alone requires an energy input of about 1100 BTU per pound. It addition, energy is required to pre-heat the wet cloth at a rate of 1 BTU per pound, per degree F. of water. The energy required for pre-heating and evaporation can be calculated by measuring the amount of moisture removed in a period of time and the initial and final temperatures of the cloth.

Energy required for heating of make-up air.

To prevent humidity from building up in the oven, the hot drying air must be exhausted. Make-up air must be added to replace that lost through the exhaust. After mixing with the existing air in the oven, the make-up air is heated up from ambient to the working temperature of the oven. After being circulated through the air handling units, this air is further heated up to 250 degrees F. The energy required for heating the make-up air can be calculated by measuring the ambient temperature, and the flow rate of this air.

Heat loss by conduction through walls of the dryer oven.

This heat loss can be calculated by determining the heat transfer coefficient of the insulation material, the oven temperature and the ambient temperature.

Total power consumption.

Efficiency of steam heating must be determined in order to calculate the total power consumption per day. Cost of energy required to run the system and cost of heat losses through the exhaust can all be calculated when the system is being operated under different ambient conditions.

Calculation of energy required for pre-heating and evaporation of moisture.

w = the removal rate of moisture in lb./min.

he = the latent heat of evaporation of water = 1100 Btu/lb.

Sw = the specific heat of water = 1 Btu/lb./degrees F.

T1 = initial temperature of moisture in degrees F.

T2 = final temperature of moisture in degrees F. (oven temperature)

Qe = energy required for evaporation in Btu/hr.

Qp = energy required for pre-heating in Btu/hr.

$Qe = w \times he \times 60$

$Qp = w \times SW \times (T2 - T1) \times 60$

TOTAL ENERGY $E1 = Qe + Qp$.

Calculation of energy required for heating of make-up air.

Sa = Specific heat of air

= 0.237 Btu/lb./degrees F.

p = Density of air

= 0.080 lb./cu. ft.

m = flow rate of make-up air in CFM.

T0 = ambient temperature of air in degrees F.

T2 = temperature of oven in degrees F.

T3 = temperature of hot air from nozzle in degrees F.

Q1 = heat required to raise the temperature from T0 to T2 degrees F. in Btu/hr.

Q2 = heat required to raise the temperature from T2 to T3 degrees F. in Btu/hr.

$Q1 = m \times p \times Sa \times 60 \times (T2 - T0)$

$Q2 = m \times p \times Sa \times 60 \times (T3 - T2)$

TOTAL ENERGY $E2 = Q1 + Q2$

Calculation of heat loss through walls of oven.

Oven is insulated by 3" fibre-glass

U = Overall heat transfer coefficient

= 3 Btu/hr. sq. ft. degrees F.

T4 = indoor temperature in degrees F.

T2 = temperature of oven in degrees F.

A = TOTAL SURFACE AREA OF OVEN IN SQ. FT.

Heat loss through walls

L = U x A x (T2 - T4) Btu/hr.

AREA OF:

END WALLS = 12.42 x 6.83 x 2 = 169.66

LARGE SIDE WALLS = 45.5 x 6.83 x 2 = 621.53

ROOF-TOP = 45.5 x 12.42 = 565.11

ROOF-TOP END WALLS = 3.5 x 12.42 x 4 = 173.88

ROOF-TOP SIDE WALLS = 3.5 x 7.5 x 4 = 105.00

TOTAL AREA A = 1635.18 SQ. FT.

Calculation of power consumption.

e = efficiency of gas heating

= 0.75

c = cost of gas = 0.4 cent per cu. ft.

H = heat generated by burning 1 cu. ft. of gas

= 1040 Btu/cu. ft.

P = power required in Btu/hr.

$$P = \frac{E1 + E2 + L}{e}$$

COST OF ENERGY PER HR.

$$C = c \times \frac{P}{H} \times 10^{-2} \text{ dollars.}$$

2.5 Data - Physical Evaluation of the Existing System

SUMMER CONDITION

w = 7.5 lb./min.

M = 30000 CFM

T0 = 75 degrees F. (DESIGN TEMP.) (24 degrees C.)

T1 = 150 degrees F. (65 degrees C.)

T2 = 220 degrees F. (104 degrees C.)

T3 = 250 degrees F. (121 degrees C.)

T4 = 75 degrees F. (24 degrees C.)

WINTER CONDITION

w = 7.5 lb./min.

m = 30000 CFM

T0 = 15 degrees F. (-12 degrees C.) (DESIGN TEMP.)

T1 = 150 degrees F. (65 degrees C.)

T2 = 220 degrees F. (104 degrees C.)

T3 = 250 degrees F. (121 degrees C.)

T4 = 75 degrees F. (24 degrees C.)

2.6 Results - Physical Evaluation of the Existing System

SUMMER CONDITION

ENERGY REQUIRED FOR EVALUATION OF MOISTURE

$Qe = w \times he \times 60$

$= 7.5 \times 1100 \times 60$

$= 495000$ Btu/hr.

ENERGY REQUIRED FOR PRE-HEATING

$Qp = w \times Sw \times (T2 - T1) \times 60$

$= 7.5 \times 1 \times (220 - 150) \times 60$

$= 31500$ Btu/hr.

TOTAL ENERGY $E1 = Qe + Qp$

$= 495000 + 31500$

$= 526500$ Btu/hr.

HEAT REQUIRED TO RAISE TEMPERATURE OF MAKE-UP AIR FROM TO TO T2

$Q1 = m \times p \times Sa \times 60 \times (T2 - TO)$

$= 30000 \times 0.080 \times 0.237 \times 60 \ (220 - 75)$

$= 4948560$ Btu/hr.

HEAT REQUIRED TO RAISE TEMPERATURE OF MAKE-UP AIR FROM T2 TO T3

$Q2 = m \times p \times Sa \times 60 \times (T3 - T2)$

$= 30000 \times 0.08 \times 0.237 \times 60 \ (250 - 220)$

$= 1023840$ Btu/hr.

TOTAL ENERGY $E2 = Q1 + Q2$

$= 4948560 + 1023840$

$= 5972400$ Btu/hr.

<u>Calculation of heat loss through walls.</u>

$L = U \times A \ (T2 - T4)$

$= 3 \times 1635.18 \ (220 - 75)$

$= 711303$ Btu/hr.

Calculation of power consumption.

$$P = \frac{E1 + E2 + 1}{e}$$

$$= \frac{526500 + 5972400 + 711303}{0.75}$$

$$= 9613604 \text{ Btu/hr.}$$

Cost of energy per hour.

SUMMER CONDITION

$$C = c \times \frac{P}{H} \times 10^{-2}$$

$$= 0.4 \times \frac{9613604}{1040} \times 10^{-2}d$$

$$= 36.98 \text{ dollars}$$

WINTER CONDITION

$$Qe = w \times he \times 60$$

$$= 7.5 \times 1100 \times 60$$

$$= 495000 \text{ Btu/hr.}$$

$$Qp = w \times Sw \times (T2 - T1) \times 60$$

$$= 7.5 \times 1 \times (220 - 150) \times 60$$

$$= 31500 \text{ Btu/hr.}$$

TOTAL ENERGY E1 = Qe + Qp

= 526500 Btu/hr.

Q1 = m x p x Sa (T2 - T1) x 60

= 30000 x 0.08 x 0.237 (220 - 15) x 60

= 6996240 Btu/hr.

Q2 = m x p x Sa x (T3 - T2) x 60

= 30000 x 0.08 x 0.237 (250 - 220) x 60

= 1023840 Btu/hr.

TOTAL ENERGY E2 = Q1 + Q2

= 8020080 Btu/hr.

HEAT LOSS

L = U x A x (T2 - T4)

= 3 x 1635.18 (220 - 75)

= 711303 Btu/hr.

$$P = \frac{E1 + E2 + L}{e}$$

$$= \frac{526500 + 8020080 + 711303}{0.75}$$

= 12,343.844 Btu/hr.

COST OF ENERGY PER HOUR

WINTER CONDITION

$$C = c \times \frac{P}{H} \times 10^{-2}$$

$$= 0.4 \times \frac{123,43844}{1040} \times 10^{-2}$$

$$= \underline{47.48 \text{ dollars}}$$

3.0 LITERATURE RESEARCH

Since 1974, with the rise in energy costs, energy saving measures have been increasingly introduced into industry all over the world.

Good housekeeping, improved insulation, better controls, condensate return and direct heat recovery are clearly among the first things to be done, but they can only do so much. The heat pump is unique in that it can recover heat from a low temperature and give out that heat at a higher and useful temperature. For many industrial processes the heat pump is the only effective means of further reducing costs.

The heat pump is not a panacea for energy saving, however; when copious quantities of air at high exhaust temperature are available it becomes a panacea in economic terms. For the heat pump to be an economic proposition there are a number of basic conditions that must be satisfied:

(1) the waste heat is available at a temperature not much lower than required; this is because the performance of a heat pump improves as its temperature range or lift is reduced.

(2) there must be no other high grade waste heat available to satisfy demand; if there is any then it will always be cheaper to use conventional heat exchangers rather than heat pumps.

(3) the heat must be required at a relatively low temperature; the present equipment is limited by the temperature at which it can deliver heat; this effectively limits operation to less than 120 degrees C. (250 degrees F.).

(4) the heat pump must have a high duty cycle; this is because it is a costly device and it will only pay if its utilization is high.

In short, because of their capital cost and technical limitations heat pumps are attractive only where they can operate over long periods, within a temperature range of 30 - 120 degrees C. (85 - 250 degrees F.), and where simpler and cheaper energy management and heat recovery techniques cannot be applied. However within these conditions there are a large number of heat pump applications to be found in industrial processes.

Substantial amounts of energy are used within industry in a variety of processes and by a variety of methods but with one aim, namely to physically remove water from the product. It has been estimated [1] that of the total energy consumed by British Industry, other than in the Iron and Steel sector, approximately 6% is used simply to vaporize water. In 1979 the total energy used by these "other industries" has been given [2] as 1920,000000 GJ and therefore about 115,000000 GJ (6% of total energy is being used solely in drying).

Products are normally dried by either one of or a combination at convection, conduction, radiation and vacuum. It is within the area of convection dryers where much work has been done to increase drying efficiency particularly by the application of heat pump dehumidifiers. In a convection drying process the product is dried by the passage of pre-heated air across its surface, the rate of drying being related to air velocity, air temperature and humidity.

One of the most successful applications of dehumidifiers is in the seasoning of timber, where the wood is "kiln dried" by following a pre-determined drying schedule in which the severity of the air condition is gradually increased as the wood moisture content decreases. Careful control is required to prevent or reduce timber degrade to within acceptable limits.

A second and more recent area of application is within the clay products industry where a wide range of products have to be dried before a high temperature firing process.

Many other industrial sectors have recently become more energy conscious and the heat pump dehumidifier is now used in a large variety of industries including:

- textiles both natural and synthetic
- leather
- gypsum
- carpet
- confectionery
- refractories.

Textiles - Wool Drying.

In the production of woollen berets, with escalating costs of fuel oil together with highly inefficient drying chambers it was decided to change completely to heat pump dehumidifier drying. The hats together with yarn in the form of cheeses, hanks and ribbons all need to be dried following the dye-house process. Previously these were dried in two oil fired fan assisted steam heated ovens at a temperature ~ 50 degrees C. (~ 125 degrees F.). Batch drying times were in the order of 5-3/4 hours which necessitated the periodic use of a third oven to relieve bottlenecks in the drying process.

Trials were conducted which proved that the heat pump dehumidifier could improve drying times and substantially reduce energy usage and drying costs.

A comparison was made between the energy usage and running cost situation of the dehumidifiers, and that of the previous heat and vent dryers. As can be seen from [3], there have

been reductions in on site energy use of approximately 90%, in primary energy use of an average 80% and in drying cost of 70% to 75%. Shorter drying time has also removed the bottleneck situation and the product quality, on a subjective basis, is said to be improved.

Based upon the total cost of converting the chambers including additional insulation, the pay back period on capital invested has been estimated as 2.4 years at 1980 price levels.

Comparisons between the performance of the oil fired system and that of the heat pump dehumidifier system are as follows:

(a)	CYCLE TIME FOR OIL FIRED SYSTEM	= 5.75 hrs.
	CYCLE TIME FOR HEAT PUMP DEHUMIDIFIER SYSTEM	= 4.5 hrs.
(b)	ON SITE SPECIFIC ENERGY CONSUMPTION FOR OIL FIRED SYSTEM	= 24080 KJ/KG
	ON SITE SPECIFIC ENERGY CONSUMPTION FOR HEAT PUMP DEHUMIDIFIER SYSTEM	= 2052 KJ/KG
(c)	PRIMARY SPECIFIC ENERGY CONSUMPTION FOR OIL FIRED SYSTEM	= 31748 KJ/KG
	PRIMARY SPECIFIC ENERGY CONSUMPTION FOR HEAT PUMP DEHUMIDIFIED SYSTEM	= 7600 KJ/KG

Clay product - gypsum mould drying.

Ceramic tableware is often produced by pouring a liquid clay slip onto or into a gypsum mould. Excess slip is removed from the mould by a simple pouring action or by a centrifuge and water from the remaining clay is rapidly absorbed by the gypsum mould. After a short period of time the clay becomes "leather hard" when sufficient water has been taken up and the ware is separated from the mould. Both the ware and the mould now need drying although in most cases the plaster mould can be used several times. Typically moulds are dried from 30% to

less than 1% moisture content. Trials were undertaken with a small 3 KW package heat pump dehumidifier which proved that the system could give high product quality at a much reduced energy consumption.

A comparison was made between the performance of the oil fired system and that of the heat pump dehumidifier system. The specific energy consumption of the oil fired system was estimated at 4600 KJ/KG on site and 7050 KJ/KG primary. A factor was included for electricity used at about 400 KJ/KG.

The specific energy consumption of the heat pump dehumidifier system was found to be 1676 KJ/KG on site and 6207 KJ/KG primary.

This represents a 64% reduction in energy usage on site and a 12% reduction in primary energy by switching to a heat pump dehumidifier system.

TIMBER DRYING

A trial was undertaken to dry identical loads of timber both by dehumidifier and in an oil fired heat and vent kiln. A load of beech was dried at the same time and over the same schedule at the kilner's site with energy consumption being closely monitored. A comparison was made between the performance of the oil fired system and that of the heat pump dehumidifier system. The specific energy consumption of the oil fired system was found to be 12294 KJ/KG on site, and 15794 KJ/KG primary. The specific energy consumption of the heat pump dehumidifier system was found to be 3477 KJ/KG on site, and 12878 KG/KG primary. The results show that for these two loads over the same moisture content reduction the heat pump dehumidifier system used only 29% of the specific energy

consumed by the heat and vent kiln per unit volume of timber. Translated into energy costs this equates to a saving of 39% based upon fuel costs applying at the time of the experiment and to a reduction in primary energy usage of 18%.

FOOD INDUSTRY

Heat pumps require a steady source of warmth, and a useful outlet for the hot water produced. The food industry rejects a great deal of heat as warm water but recovery is only worthwhile if the temperature can be brought up close to 100 degrees C. A research into selected food factories shows that there is not much potential use for the hot water. Evaporation of water is an important part of many food processes. In some factories over 40% of the energy input is used for evaporation.

There are a number of techniques available for recovering heat from evaporated vapors, in the food industry the problem is more complex because very often the vapors will carry fine food particles which will foul heat transfer surfaces, will get sticky when moist and provide a potential breeding ground for bacteria. The risks of modifying a food plant to include heat recovery are considerable, particularly in higher seasonal sectors of the business, like milk drying. More search is required to demonstrate the feasibility of heat pump under practical conditions.

Also the food industry is a large user of metric devices for refrigeration equipment, in addition to that for processing and packing machines. Electrical motors are relatively inexpensive in capital terms but expensive in energy terms because a premium energy is being used. It is quite possible to use recovered heat to drive these refrigeration plants, though at present the capital costs appear to be too high, but there is a prospect of developing an economic solution to this problem.

4.0 THEORY

The term drying refers generally to the removal of moisture from a substance. The moisture contained in a wet solid or liquid solution exerts a vapor pressure to an extent depending upon the nature of the moisture, the nature of the solid, and the temperature. If then a wet solid is exposed to a continuous supply of fresh gas containing a fixed partial pressure of the vapor $\bar{p}$, the solid will either lose moisture by evaporation or gain moisture from the gas until the vapor pressure of the moisture of the solid equals $\bar{p}$.

Drying operations can be broadly classified according to whether they are batch or continuous. These terms are applied specifically from the point of view of the substance being dried. Thus the operation termed batch drying is usually in fact a semi-batch process wherein a quantity of the substance to be dried is exposed to a continuous flowing stream of air into which the moisture evaporates.

4.1 Method of Supplying the Heat Necessary for Evaporation of the Moisture

In direct driers, the heat is supplied entirely by direct contact of the substance with the hot gas into which evaporation takes place.

In indirect driers, the heat is supplied quite independently of the gas used to carry away the vaporized moisture.

It is a very important point that in convectional type dryers the temperature of the drying material has a wet-bulb temperature. In direct driers the temperature of the moisture is

close to the temperature of hot surface and correspondently partial pressure of the vapor is higher.

Drying efficiency is a term which has many variants but in its most useful form it is a measure of the quantity of energy expended in removing unit mass of water from a product. Normally this is measure in terms of KJ/KG although when considering systems incorporating electrically driven heat pumps units of KWH/Kg are more readily useful. In the conventional heat and vent dryer the thermal efficiency can be increased by reducing the ventilation rate.

The efficiency can be written by the approximation [4]

$$\eta f = \frac{T1 - T2}{(T1 - T2) + (1-V) + (T1 - T2)}$$

where

ηf - thermal efficiency
T1 - temperature of air after passing over heating coil.
T2 - temperature of exhaust air.
V - ratio of recirculated air to total air flow within dryer.

If the rate of ventilation is reduced, the ratio "V" of recirculated air to total air flow will increase and therefore the efficiency will increase. This, however, will tend to increase drying times as the process moisture removal rate will fall. Although in theory, it is possible for the thermal efficiency of a convection dryer to approach 100% by reducing the ventilation rate to zero, in practice there must be a reduction in thermal efficiency.

A useful yardstick when comparing drying processes is the latent heat of vaporization of water at ambient temperature.

This figure is approximately 2300 KJ/KG and serves as an indication of the efficiency of a convection dryer operating at 100% efficiency.

4.2 Rate of Drying

In our case the section of cloth is dried in a stream of gas. The whole part is immersed in a stream of drying gas at temperature T_G and humidity Y - mass moisture/mass dry gas, flowing at mass velocity G - mass/ (time) (area). The evaporation of moisture takes place from the upper surface area A, at a temperature Ts. The drying cloth receives heat from direct contact with hot surface qk by conduction, qr by direct radiation from a hot duct-surface at temperature T_R and q_c by convection from the gas stream.

The rate of evaporation and the surface temperature can then be obtained by a heat balance.

$$q = q_c + q_k + q_r$$

We can neglect the heat required to super heat the evaporated moisture to the gas temperature and consider only the latent heat of vaporization λ_s, the flux of evaporation N_c and the flux of heat flow is:

$$N_c \lambda_s = q \quad (1)$$

The heat received at the surface by convection is controlled by the convection heat - transfer coefficient h_c:

$$q_c = h_c (T_g - T_s) \quad (2)$$

The heat received by radiation can be estimated by the equation:

$$q_r = h_r (T_r - T_s) \quad (3)$$

where

$$h_r = \frac{E (5.729 \times 10^{-8}) (T_r^4 - T_s^4)}{T_r - T_s} \quad (4)$$

E - emmissivity of drying surface

The heat received by convection and conduction can be computed:

$$q_k k = U_k (T_g - T_s) \quad (5)$$

U_k - overall heat-transfer coefficient

Combining Eqs (1-5) permits calculation of the rate of drying,

$$N_c = \frac{q}{\lambda_s} = \frac{(ho + V_k) (Tg - Ts) + hr (Tr - Ts)}{\lambda_s} = Ky (Y_s - Y) \quad (6)$$

where

Ky = gas - phase mass - transfer coefficient

The rate of drying should be considered as function of several parameters as: gas velocity, gas temperature, gas humidity and thickness of drying cloth.

Effect of gas velocity.

If radiation and conduction through the cloth are negligible, N_c is proportional to $G^{0.71}$ for parallel flow of gas and to $G^{0.37}$ for perpendicular flow (for G = 1 -: 5 kg/m^2 x s)

If radiation and conduction are present, the effect of gas rate will be less important.

Effect of gas temperature.

Increase of air temperature increases the quantity $(T_g - T_s)$ and hence increases N_c. In the absence of radiation effects, if the variation of λ_s over moderate temperature ranges is neglected, N_c is directly proportional to $(T_g - T_s)$.

Effect of gas humidity.

N_c varies directly with Ys - Y, and consequently increasing the humidity lowers the rate of drying. Usually, changes in Y and T_g involve simultaneous changes in T_s and Y_s, and the effects are best estimated by direct application of E q (6).

Effect of thickness of drying solid.

If heat conduction through the cloth occurs, E q (5) indicates lowered values of N_c with increased cloth thickness. The time for drying between fixed moisture contents within the constant-rate period will then be directly proportional to thickness.

We can see that the evaporating and condensing temperatures are very important for evaporating rate. Gas humidity is also a function of the evaporating temperature. Decrease of vapor temperature will increase the dehumidification efficiency of

the heat pump but at the same time it will reduce compressor capacity and will require more horse-power per ton of refrigeration.

The highest efficiency of heat pump dehumidifier can be achieved by optimization of evaporating temperature.

Dehumidification can be shown on Mollier chart by the straight line that connects the two points, i.e. point A - condition of the air around the dehumidifier, and point B - saturated air at cooling surface temperature.

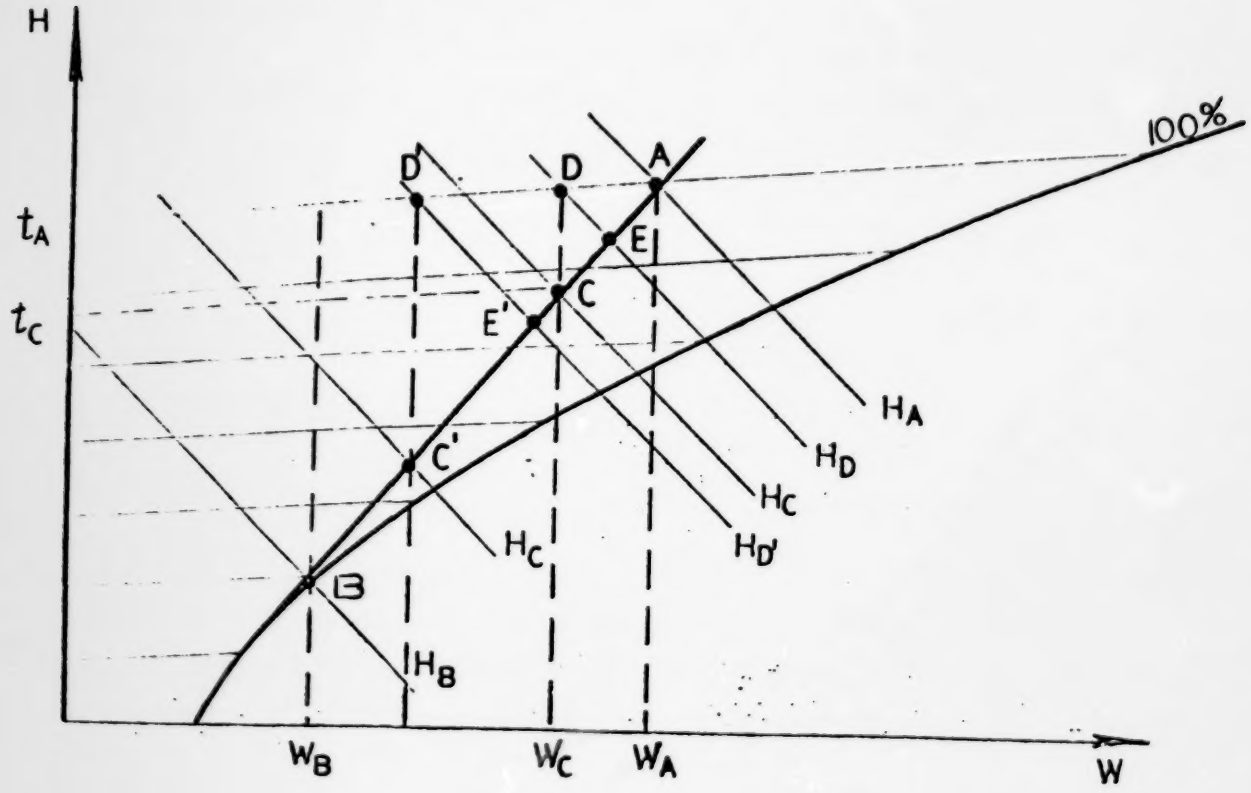


fig. 1

A change of state from A to C, requires a cooling surface effect equal to (H_A - H_c) K-cal per Kg of air. This effect is made up of a latent load: $Q_L = H_A - H_D$, employed to

precipitate the liquid quantity $X_A - X_C$ Kg water/ Kg air, and a sensible load: $Q_s = H_D - H_C$ for the cooling down of every Kg air from T_A to T_C. The optimization of the dehumidification process, we are particularly interested in the ratio between latent and sensible loads:

$$\frac{QL}{Q_S} = \frac{H_A - H_D}{HD - HC}$$

or

$$\frac{Q_L}{Q_S} \quad \frac{AE}{EC}$$

If, instead, the state of the air had actually been the point C', we would correspondingly:

$$\frac{QL}{QS} = \frac{AE'}{E'C'}$$

Since the triangles ADC and AD'C' are similar the following applies:

$$\frac{QL}{QS} = \frac{AE}{EC} = \frac{AE^1}{E^1C^1}$$

Whereby we have shown, that the state of the air around the dehumidifier and the cooling surface temperature alone determines the ratio between latent and sensible loads.

OPTIMIZATION of the ratio $\frac{QL}{QS}$

This can be done graphically by means of the Mollier chart. An analytical solution would, however, be more expedient at this point, and for this we apply "Backstrom's" equation.

$$\frac{QL}{QS} = K \times \frac{Pwa - Psws}{t_z - t_s}$$

Pwa - is the partial pressure of the water vapors in the state of air surrounding the dehumidifier.

Psws - is the partial pressure of the water vapours in vapour saturated air of cooling element surface temperature.

Ta - the temperature of the air surrounding the dehumidifier.

ts - the cooling element surface temperature.

K - a constant.

By means of "Backstrom's" equation we can draw the curve $\frac{QL}{QS}$, as a function of the cooling element surface temperature at an arbitrarily chosen state of the air surrounding the dehumidifier.

N	t_s	Pwa - Psws	t_a - ts	QL/Q_S
1	110 degrees F.	0.33	44	0.7
2	100 degrees F.	0.99	54	1.7
3	90 degrees F.	1.5	64	2.19
4	80 degrees F.	1.88	74	2.37
5	70 degrees F.	2.17	84	2.4

fig. 2

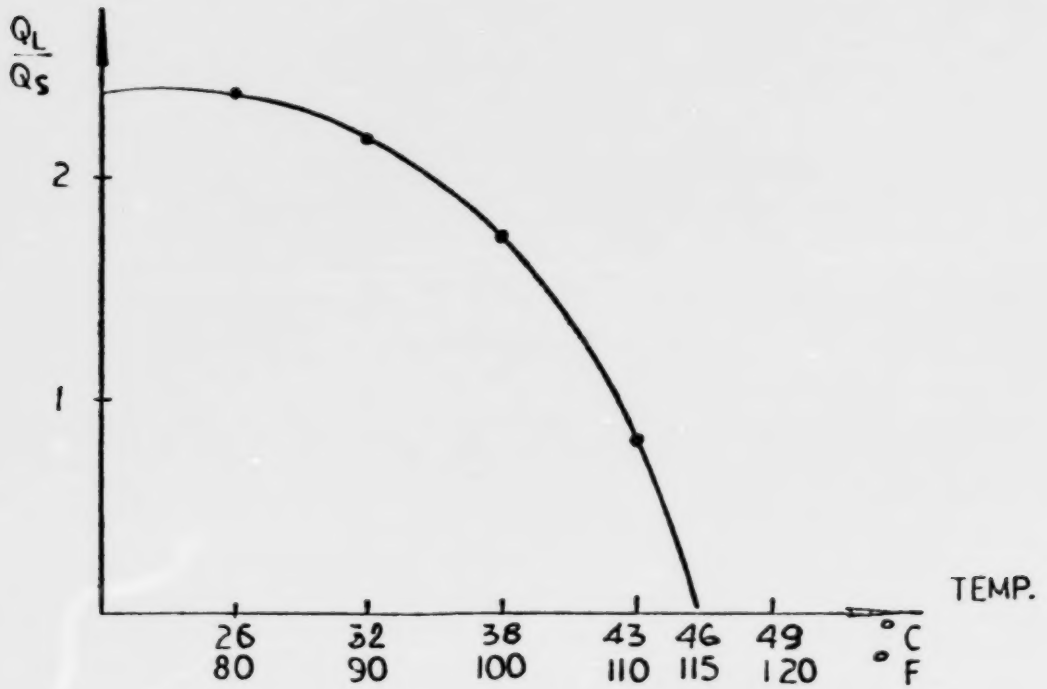


fig. 3

From fig. 3 it appears that $\frac{QL}{Q_s}$ shows maximum at approximately 26 degrees C. (80 degrees F.) and minimum at 46 degrees C. (115 degrees F.) cooling element surface temperatures.

In the Mollier chart this corresponds, both to Maximum gradient of the line surrounding state/surface temperature on curve of 100% humidity.

We know that optimum of $\frac{QL}{Q_s}$ at an evaporator surface temperature is approximately 26 degrees C. However, we should also be obliged to consider that compressors capacity is dependent on the suction pressure, and thus on the evaporator surface temperature.

Fig. 4 shows compressor capacity as function of surface temperature. The curve is otherwise drawn for constant condensing temperature.

By means of fig. 3 we divide the compressor capacity into a sensible and latent share, both of which are shown on fig. 4.

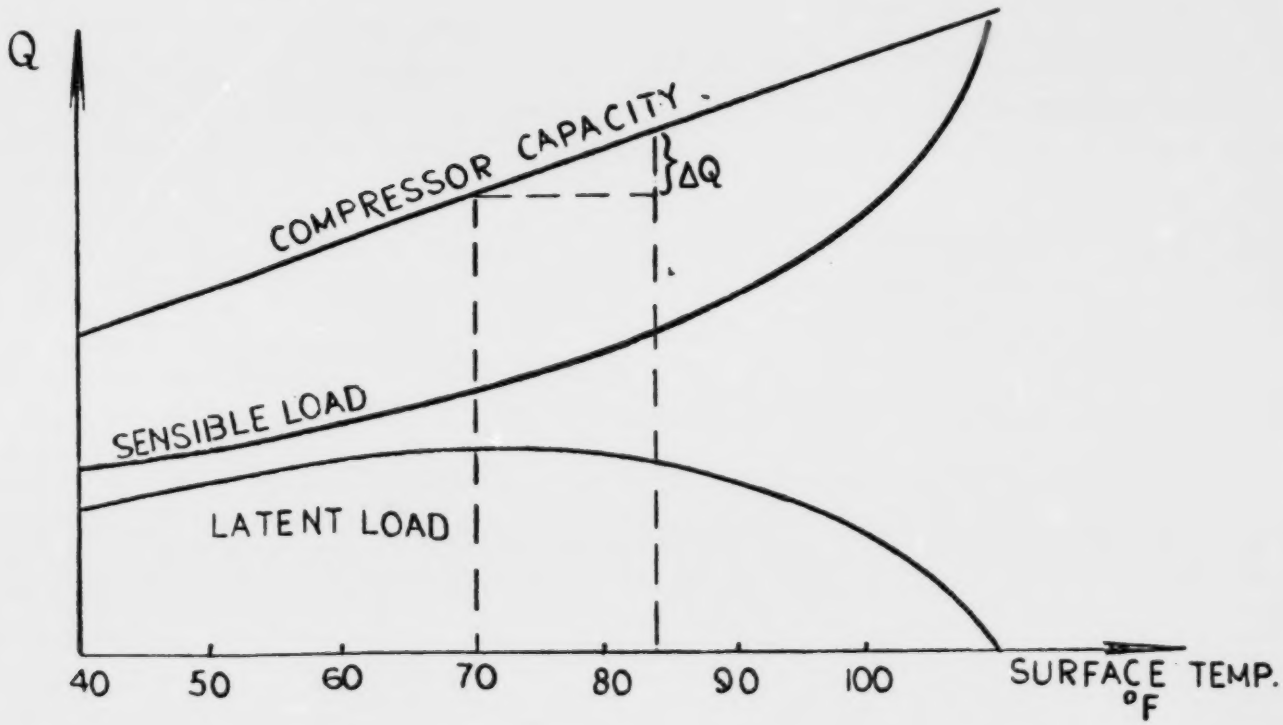


fig. 4

With dehumidifiers interest centres primarily around the latent load curve as a function of the evaporator surface temperature. This curve shows a markedly flat maximum within the surface temperature range ≈ 70 degrees F. (21 degrees C.) - 83 degrees F. (28 degrees C.), it might be considered immaterial, where the surface temperature was found within this compass.

Fig. 4 is, however, drawn on the basis of the rather theoretical assumption that the condensing temperature is kept constant by varying the evaporator surface temperature from 83 degrees F. (28 degrees C.) to 70 degreee F. (21 degrees C.), accompanied by a diminishing of the evaporator's surface area. Since the compressor's capacity is reduced by ΔQ (fig. 4) the quantity of heat to be removed in the condenser is reduced and Q simultaneous decrease of the condenser area will thus be required to keep the condensing temperature constant. If the evaporator alone is modified to effect the surface temperature drop from 83 degrees F. (28 degrees C.) to 70 degrees F. (21 degrees C.), the said condenser would have to remove less heat, and a lower condensing temperature would result. The compressor's capacity will increase along with the drop in the condensing temperature. The system will thus find an equilibrium, with a compressor capacity answering to the new condensing temperature.

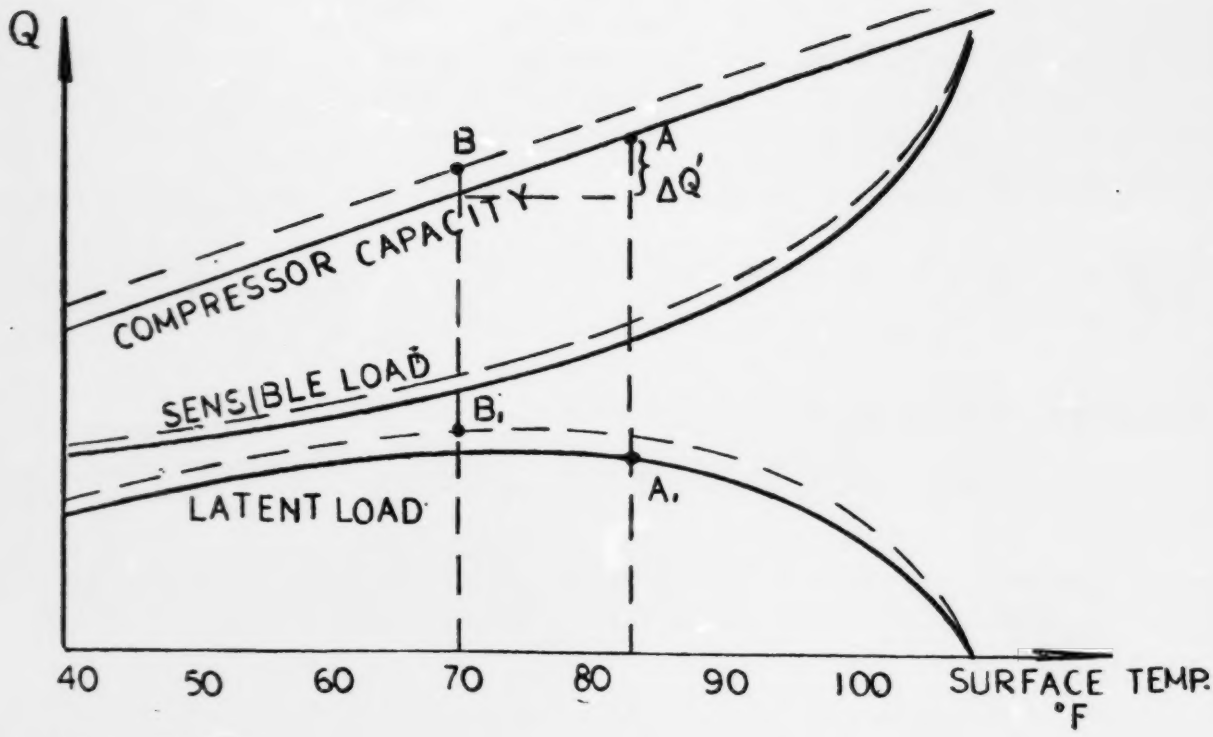


fig. 5

——— CAPACITY AT CONDENSING TEMP. = T

—.— CAPACITY AT CONDENSING TEMP. = T + ΔT

Fig. 5 repeats the capacity curves of fig. 4, with two additional condensing temperatures drawn in T and (T - ΔT), where dT stands for the lowering the the condensing temperature, due to the compressor capacity change from A to B, (ΔQ'). A lowering of the surface temperature, brought about solely by altering the evaporator surface temperature, will consequently lead to a small improvement of the latent load, from A, to B, as well as a saving in material costs.

It has thus been shown, that the optimized operating state of a humidifier, will be found at the lowest end of the 70 degree F. (21 degree C.) to 83 degrees F. (28 degrees C.) compass.

The recording made from mathematical modelling of the system determined optimum design parameters, as evaporating temperature at 80 degrees F. and condensing temperature at 190 degrees F.

4.3 Refrigerant Selection

The condensing temperature chosen restricts the selection of refrigerants to those which have a critical temperature in excess of 120 degrees C. (~ 250 degrees F.), ruling out commonly used fluids such as R12 and R22. In order to reduce the choice further, the properties of refrigerants which are most desirable in the application must be addressed.

In this context thermal stability is particularly important. While breakdown can be tolerated, refrigerants are expensive and thus regular replacement is undesirable. Also, acids can be formed by breakdown, leading to corrosion in the compressor and pipework. Breakdown is accelerated at high temperature, and is also a function of the materials which the refrigerant is brought into contact.

Refrigerant 114 appears to be consistently much more stable than other candidates, because of the high proportion of fluorine contained in the molecule. As fluorine is a strongly electro-negative element, the bonds within the refrigerant molecule become polarised and therefore much stronger, thus making the molecule more stable. Consideration of the calculated discharge temperature revealed that in all cases they were above the ASHRAE recommended maximum continuous operating temperatures, but R 114 would be operating closest to its maximum recommended temperature at 125 degrees C. (~ 260 degrees F.).

R 113 appeared to be the next most stable refrigerant, following by R 11, and R 21 was the least stable of the four which are mentioned above.

R 113, the next after R 114 best fluid with a discharge temperature at 113 degrees C. (~ 232 degrees F.), would be operating at 26 degrees C. (80 degrees F.) above its maximum continuous operating temperature of 107 degrees C. (222 degrees F.).

Narrowing the selection down to R114 and R113 an examination of volumetric flow rate shows that for R113 the value is too high for reciprocating compressors which we believe are most appropriate for the duty here, and also the pressure ratio is high. While R114 has a lower COP and higher discharge and suction pressure, with implications for thrust bearing design, its stability, lower discharge temperature, and greater suitability for reciprocating compressors led us to select it as the fluid to be used in the project.

Solubility of the refrigerant in the compressor oil is related to temperature, and of course can have a marked effect on compressor operation.

The most critical factor is the oil viscosity, which decreased with increasing refrigerant concentration, leading to bearing lubrication problems, forming acids and sludges. We select special type of oil for high temperature application.

5.0 HEAT PUMP THEORY

A heat pump is essentially a heat engine operating in reverse. From the first law of thermo-dynamics, the amount of heat delivered Q_D at the higher temperature T_D is related to the amount of heat extracted Q_s at the low temperature T_s and the amount of high grade energy input W by the equation.

$$Q_D = Q_s + W$$

A coefficient of performance (COP) can be defined as:

$$(COP) = \frac{QD}{W}$$

A heat engine operating between a higher temperature T_D and a lower - T_s has a theoretical maximum thermodynamic efficiency known as the Carnot efficiency.

$$\eta = (TD - T_s)/T_D$$

The Carnot coefficient of performance defined by equation.

$$(COP)_c = \frac{TD}{TD - Ts}$$

represents the upper theoretical value obtainable in a heat pump system. In practice obtainable coefficient of performance are significantly less that $(COP)_c$.

The difference between the condensing and evaporating temperatures (Tco - Tev) is the gross or maximum possible temperature lift. The net temperature lift ($TD-T_s$) is less than the gross temperature lift by the sum of the temperature difference

driving forces in the evaporator and condensor heat exchangers.

The ratio of the corresponding pressures in the condenser and evaporator Pco/Pev is the compression ratio (CR).

(Tco-Tev), (CR) and the condensing temperature Tco are the critical parameters which determine the feasible operating range of the heat pump operating on the particular fluid.

In practice the operation of a mechanical vapour compression heat pump approximates more closely to the Rankine heat pump cycle than the theoretical Carnot cycle [5].

A heat pump effectiveness compared to the theoretical Rankine cycle can be defined as:

$$(HPE)_R = (COP)_A \; / \; (COP)_R$$

which is the ratio at the actual coefficient of performance to the calculated ideal Rankine coefficient of performance.

The $(HPE)_R$, i.e. effectively the ratio of work required in an ideal Rankine cycle to that in an actual cycle in order to transfer a given amount of heat to the condenser.

The design parameters for heat pump systems are the compression ratio (CR), the condensation temperature Tco, the gross temperature lift (Tco-Tev) and the theoretical Rankine coefficient of performance $(COP)_R$. When two of these parameters are fixed the other two are automatically determined. If (CR) is determined by the availability of the compressor and Tco by the temperature of the required process heat, then (Tco - Tev) and $(COP)_R$ are also fixed for a particular working fluid.

5.1 Heat Pump Design

(1) The cooling duty required:

In accordance the data from Canada Sandpaper, the rate of evaporating moisture is 470 lb/hr.

Latent load on evaporater is:

$$Q_s^1 = 470 \times 1100 \times 1.15 = 594550 \text{ BTU/hr.} = 50 \text{ T.R.}$$

The dehumidification effect in evaporator 1

$$\frac{Q \text{ Latent}}{Q \text{ Total}} = \frac{1}{1 + \frac{Q \text{ sen.}}{Q \text{ Latent}}} = \frac{1}{1.42} = 0.7$$

The total heat load on the evaporator 1

$$Q_S = \frac{594550}{0.7} \times 1.2 = 1019228.6 \text{ BTU/hr.} = 299 \text{ KW} = 85 \text{ T.R.}$$

(2) The heating duty required

$$QD = Q_s + Q \text{ compressor} = 1691919.5 \text{ BTU/hr.} = 496 \text{ KW.}$$

(3) $$(COP)\ A = \frac{QD}{Q_D - Q_s} = \frac{496}{996-299} = 2.5$$

The temperature drops required in the evaporator and condenser heat exchangers, inter alia, on the areas and conformation of the heat exchangers.

Assume them to be 10 degrees C.

The values of the condensing and evaporating temperatures are
Tco = 78 degrees C. + 10 degrees C. = 88 degrees C.
= 190 degrees F. = 361 degrees K.
and Tev = 36 degrees C. - 10 degrees C. = 26 degrees C.
= 80 degrees F. = 299 degrees K.

The corresponding gross temperature lift is therefore
(Tco - Tev) = 88 degrees - 26 degrees = 62 degrees C.
The pressure Pco = 11.0 bar.

With a gross temperature life of 62 degrees C. the corresponding compression rate (CR) = 4.98 [6]

From [6] the Rankine coefficient of performance
$(COP)_A$ = 3.85 for R114 at Tco = 88 degrees C. and
(Tc0 - Tec) = 62 degrees C.

The heat pump effectiveness factor $(HPE)_R$ is approximately 0.85. The coefficient of performance to be expected from the system is therefore:

$$(COP)_A = 0.85 \times 3.85 = \underline{\underline{3.27}}$$

In many cases the maximum compression ratio achievable by a given compressor will be a limiting factor for the system. Also there will be pressure drops from the compressor outlet to the expansion valve, and from expansion valve to the compressor which, unlike the pressure drop across the expansion valve, are not adiabatic. However, these departures from ideality, as well as the effects of inefficiencies in the compressor itself are included in the experimental valves of $(HPE)_R$ = 0.85.

Duty limited by the compression ratio of the compressor.

The isentropic efficiency of the compressor may be determined from the expression [7]:

I.E. ≅ 1 - 0.05 (CR)

In practice, the efficiency will usually decrease rapidly if the maximum recommended compression ratio is exceeded.

Compressor has a limiting ratio of 7 at which its mechanical efficiency is 0.92.

I.E. ≅ 1-0.05 x 7 = 0.65

Hence, the overall efficiency of the compressor:

0.65 x 0.92 = 0.6

This implies that, at the power supplied to the compressor, only 60 percent is available to carry out the rankine cycle, the remainder is converted into heat.

Examination of derived thermodynamic design data for heat pump systems operating on R114 show that the given compression ratio can be achieved for a range of condensing temperatures and gross temperature lifts.

Suppose that a temperature drop of 3 degrees C. in each exchanger.

Tco = 88 degrees C. + 3 degrees C. = 91 degrees C.

Tco - Tev = 91 - (26 degrees C. - 3 degrees C.) = 68 degrees C

In the present case, compression ratio for R114 is CR = 5.89

Since Pco = 11.71 bar at 91 degrees C., Pev = 11.71/5.89
= 1.988 bar

Assume a pressure drop from compressor to condenser 0.5 and between expansion valve and the compressor inlet 0.1.

The actual compression ratio:

11.71 + 0.5/1.988 - 0.1 = 6.467

This choice of condensing and evaporating temperatures is, therefore, satisfactory both from the point of view of the compression ratio and of the maximum operating pressure of the system.

Because of the pressure drops in the heat exchangers the mean condensing and evaporating temperatures will be higher than those estimated above.

The mean pressure in the condenser
11.71 + 0.5/2 = 11.96 bar

The mean saturation temperature in the condenser

$$91 + \frac{(11.96 - 11.71)}{(11.957 - 11.71)} \times 1 \text{ degree C.} = 92 \text{ degrees C.}$$

The mean pressure in the evaporator = 1.988 + 0.1/2 = 2.238

Tev = 26.9 degrees C.

Tco - Tev = 92 degrees C. - 25.9 degrees C. = 65 degrees C.
(CR) = 5.26
$(COP)_R$ = 3.58

Now estimate approximately the Rankine heat pump effectiveness factor $(HPE)_R$

Tco = 82 degrees C.

Tco - Tev = 53 degrees C.

$(COP)_R = 4.77$

$(CR)_R = 4.01$

Therefore, $(HPE)_R \cong \frac{3.58}{4.77} = 0.75$ for mechanically perfect compressor.

Since the compressor is only 92 percent efficient, $(HPE)_R \cong 0.75 \times 0.92 = 0.69$ is in reasonable agreement with value 0.6

$(COP)_A \cong 4.77 \times 0.69 = 3.29$

The required heat rate from condenser $Q_D = 496$ KW

The work to be provided at the shaft of the compressor is therefore:

$$W \cong \frac{496}{3.29} = 150.7KW$$

The energy to be removed from the heat source

$$Q_S = Q_D - W = 496 - 150.7 = 345.3KW$$

This is in excess of the cold actually required by 345.3 - 299 = 46.3KW

To match both the heating and cooling requirements we can by adjustment of the temperature driving forces in the heat exchanger.

To match the required heat rate:

$(COP)_A = Q_D/(Q_D - Q_S) = 496/(496 - 299) = 2.5$
$(COP)_R = 2.5/0.69 = 3.6$

DERIVED THERMODYNAMIC DATA FOR DESIGN.

evaporator exit temperature	25 degrees C. (77 degrees F.)
compressor inlet temperature	45 degrees C. (116 degrees F.)
compressor outlet temperature	89 degrees C. (192 degrees F.)
condenser outlet temperature	87 degrees C. (188 degrees F.)
expansion valve inlet temperature	34.7 degrees C.(94 degrees F.)
condenser duty	496KW
evaporator duty	299KW
pressure of compressor inlet	12 bar
pressure of compressor outlet	2.28 bar
compression ratio of compressor	5.26
heat pump effectiveness	0.75
coefficient of performance	2.5

T_{CO} °C (P_{CO} bar)	15	16	17	18	19	20	21	22	23	24	25
$(T_{CO} - T_{EV})$ °C	1.5193	1.5734	1.6289	1.6860	1.7496	1.8047	1.8664	1.9297	1.9946	2.0612	2.1295
10	27.58	27.67	27.76	27.84	27.91	27.99	28.06	28.15	28.22	28.32	28.39
15	18.07	18.11	18.18	18.22	18.27	18.32	18.37	18.42	18.47	18.52	18.57
20	13.32	13.35	13.39	13.42	13.46	13.49	13.53	13.57	13.60	13.63	13.67
25	10.47	10.50	10.53	10.55	10.58	10.61	10.63	10.66	10.68	10.71	10.73
30	8.58	8.60	8.62	8.64	8.66	8.68	8.70	8.72	8.74	8.76	8.78
35	7.23	7.25	7.27	7.28	7.30	7.31	7.33	7.35	7.36	7.38	7.39
40	6.23	6.24	6.25	6.27	6.28	6.29	6.31	6.32	6.33	6.34	6.36
45	5.45	5.46	5.47	5.48	5.49	5.50	5.51	5.52	5.53	5.54	5.55
50	4.83	4.84	4.85	4.85	4.86	4.87	4.88	4.89	4.90	4.91	4.91
55	4.32	4.33	4.34	4.34	4.35	4.36	4.37	4.37	4.38	4.39	4.39
60	3.90	3.91	3.92	3.92	3.93	3.93	3.94	3.95	3.95	3.96	3.96
65	3.55	3.56	3.56	3.57	3.57	3.58	3.58	3.59	3.59	3.60	3.60
70	3.25	3.26	3.26	3.27	3.27	3.27	3.28	3.28	3.29	3.29	3.29
75	2.99	3.00	3.00	3.01	3.01	3.01	3.02	3.02	3.02	3.03	3.03

THEORETICAL RINKLINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar) / $(T_{CO} - T_{EV})$ °C	15	16	17	18	19	20	21	22	23	24	25
	1.5193	1.5734	1.6289	1.6860	1.7496	1.8047	1.8664	1.9297	1.9946	2.0612	2.1295
10	1.44	1.44	1.43	1.43	1.43	1.42	1.42	1.41	1.41	1.41	1.40
15	1.75	1.74	1.74	1.73	1.73	1.71	1.71	1.70	1.69	1.68	1.68
20	2.14	2.13	2.12	2.11	2.10	2.08	2.07	2.06	2.04	2.03	2.02
25	2.65	2.63	2.61	2.59	2.57	2.55	2.53	2.51	2.49	2.47	2.45
30	3.30	3.27	3.24	3.21	3.19	3.15	3.12	3.09	3.06	3.03	3.01
35	4.16	4.11	4.06	4.02	3.98	3.92	3.88	3.84	3.80	3.75	3.71
40	5.29	5.21	5.15	5.07	5.02	4.94	4.88	4.81	4.75	4.69	4.63
45	6.80	6.70	6.58	6.49	6.40	6.29	6.18	6.10	6.00	5.92	5.83
50	8.86	8.68	8.53	8.37	8.25	8.00	7.94	7.80	7.67	7.54	7.42
55	11.66	11.43	11.18	10.96	10.76	10.52	10.30	10.11	9.90	9.72	9.53
60	15.60	15.20	14.87	14.50	14.23	13.85	13.56	13.24	12.97	12.67	12.42
65	21.11	20.58	20.00	19.51	19.04	18.53	18.03	17.61	17.16	16.76	16.34
70	29.14	28.21	27.43	26.58	25.93	25.08	24.41	23.70	23.09	22.43	21.86
75	41.15	39.50	38.11	36.94	35.79	34.61	33.47	32.49	31.45	30.55	29.59

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar)	25	26	27	28	29	30	31	32	33	34	34
$(T_{CO} - T_{EV})$ °C	2,1295	2,1914	2,2711	2,3446	2,4198	2,4968	2,5757	2,6564	2,7390	2,8235	2,9100
10	28,39	28,47	28,53	28,63	28,70	28,78	28,85	28,93	29,01	29,09	29,16
15	18,57	18,61	18,66	18,72	18,77	18,81	18,86	18,90	18,95	19,00	19,05
20	13,67	13,70	13,74	13,78	13,81	13,84	13,87	13,90	13,93	13,97	14,00
25	10,73	10,76	10,78	10,81	10,84	10,86	10,88	10,91	10,93	10,96	10,98
30	8,78	8,80	8,82	8,84	8,86	8,88	8,90	8,92	8,93	8,95	8,97
35	7,39	7,41	7,42	7,44	7,46	7,47	7,48	7,50	7,51	7,53	7,54
40	6,36	6,37	6,38	6,39	6,40	6,42	6,43	6,44	6,45	6,46	6,47
45	5,55	5,56	5,57	5,58	5,59	5,60	5,61	5,62	5,63	5,64	5,65
50	4,91	4,92	4,93	4,94	4,95	4,95	4,96	4,97	4,97	4,98	4,99
55	4,39	4,40	4,41	4,41	4,42	4,42	4,43	4,44	4,44	4,45	4,45
60	3,96	3,97	3,97	3,98	3,98	3,99	3,99	4,00	4,00	4,01	4,01
65	3,60	3,61	3,61	3,61	3,62	3,62	3,63	3,63	3,63	3,64	3,64
70	3,29	3,30	3,30	3,30	3,31	3,31	3,31	3,32	3,32	3,32	3,32
75	3,03	3,03	3,04	3,04	3,04	3,04	3,04	3,05	3,05	3,05	3,05

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	25	26	27	28	29	30	31	32	33	34	34
$(T_{CO} - T_{EV})$ °C	2.1295	2.1914	2.2711	2.3446	2.4198	2.4968	2.5757	2.6564	2.7390	2.8235	2.9100
10	1.40	1.40	1.39	1.39	1.39	1.38	1.38	1.38	1.37	1.37	1.37
15	1.68	1.67	1.66	1.66	1.65	1.64	1.64	1.63	1.62	1.62	1.61
20	2.02	2.01	2.00	1.99	1.98	1.97	1.96	1,95	1.94	1.93	1.92
25	2.45	2.44	2.42	2.40	2.39	2.37	2.35	2.34	2.32	2.31	2.29
30	3.01	2.98	2.95	2.93	2.90	2.88	2.85	2.83	2.81	2.78	2.76
35	3.71	3.67	2.64	3.60	3.56	3.52	3.49	3.45	3.42	3.39	3.35
40	4.63	4.57	4.52	4.46	4.41	4.35	4.30	4.25	4.20	4.16	4.11
45	5.83	5.75	5.66	5.58	5.50	5.43	5.35	5.28	5.21	5.14	5.08
50	7.42	7.29	7.18	7.05	6.95	6.84	6.73	6.62	6.52	6.42	6.33
55	9.53	9.36	9.18	9.02	8.85	8.70	8.53	8.39	8.21	8.11	7.97
60	12.12	12.14	11.90	11.64	11.42	11.17	10.96	10.74	10.54	10.32	10.14
65	16.34	15.98	15.58	15.24	14.88	14.56	14.22	13.92	13.60	13.32	13.02
70	21.86	21.25	20.73	20.17	19.68	19.16	18.71	18.23	17.81	17.36	16.97
75	29.59	28.77	27.89	27.14	26.33	25.63	24.89	24.25	23.56	22.96	22.32

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	35	36	37	38	39	40	41	42	43	44	45
$(T_{CO}-T_{EV})$ °C	2.9100	2.9984	3.0888	3.1813	3.2758	3.3723	3.4710	3.5718	3.6747	3.7798	3.8871
10	29.16	29.23	29.30	29.35	29.42	29.51	29.60	29.67	29.73	29.78	29.86
15	19.05	19.09	19.14	19.18	19.22	19.27	19.32	19.36	19.40	19.44	19.49
20	14.00	14.03	14.06	14.09	14.12	14.16	14.19	14.22	14.25	14.28	14.30
25	10.98	11.00	11.02	11.05	11.07	11.09	11.12	11.14	11.16	11.18	11.20
30	8.97	8.99	9.00	9.02	9.04	9.06	9.07	9.09	9.11	9.12	9.14
35	7.54	7.55	7.57	7.58	7.59	7.61	7.62	7.63	7.65	7.66	7.67
40	6.47	6.48	6.49	6.50	6.51	6.52	6.53	6.54	6.55	6.56	6.57
45	5.65	5.65	5.66	5.67	5.68	5.69	5.69	5.70	5.71	5.72	5.72
50	4.99	4.99	5.00	5.01	5.01	5.02	5.03	5.03	5.04	5.04	5.05
55	4.45	4.46	4.46	4.47	4.47	4.48	4.48	4.49	4.49	4.49	4.50
60	4.01	4.01	4.02	4.02	4.03	4.03	4.03	4.04	4.04	4.04	4.04
65	3.64	3.64	3.64	3.65	3.65	3.65	3.65	3.66	3.66	3.66	3.66
70	3.32	3.33	3.33	3.33	3.33	3.33	3.33	3.34	3.34	3.34	3.34
75	3.05	3.05	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	35	36	37	38	39	40	41	42	43	44	45
$(T_{CO}-T_{EV})$ °C	2,9100	2,9984	3,0888	3,1813	3,2758	3,3723	3,4710	3,5718	3,6747	3,7798	3,8871
10	1,37	1,36	1,36	1,36	1,35	1,35	1,35	1,34	1,34	1,34	1,34
15	1,61	1,61	1,60	1,59	1,59	1,58	1,58	1,56	1,57	1,56	1,56
20	1,92	1,91	1,90	1,89	1,88	1,87	1,86	1,85	1,84	1,83	1,83
25	2,29	2,28	2,26	2,25	2,23	2,22	2,21	2,19	2,18	2,17	2,15
30	2,76	2,74	2,72	2,70	2,68	2,66	2,64	2,62	2,60	2,58	2,56
35	3,35	3,32	3,29	3,26	3,23	3,20	3,17	3,14	4,11	3,09	3,06
40	4,11	4,06	4,02	3,97	3,93	3,89	3,85	3,81	3,77	3,73	3,69
45	5,08	5,01	4,94	4,88	4,82	4,76	4,70	4,64	4,59	4,53	3,48
50	6,33	6,23	6,14	6,05	5,97	5,88	5,80	5,72	5,64	5,56	5,49
55	7,97	7,83	7,70	7,58	7,45	7,33	7,21	7,10	6,99	6,88	6,78
60	10,14	9,93	9,76	9,57	9,40	9,24	9,07	8,90	8,75	8,59	8,45
65	13,02	12,76	12,48	12,24	11,98	11,75	11,50	11,28	11,05	10,85	10,65
70	16,97	16,55	16,18	15,80	15,46	15,09	14,77	14,43	14,14	13,82	13,54
75	22,33	21,78	21,19	20,68	20,14	19,66	19,16	18,72	18,25	17,83	17,40

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	45	46	47	48	49	50	51	52	53	54	55
$(T_{CO} - T_{EV})$ °C	3.8871	3.9967	4.1085	4.2226	4.2226	4.4577	4.5788	4.7024	4.8283	4.9567	5.0876
10	29.86	29.93	29.99	30.08	30.15	30.21	30.26	30.31	30.40	30.45	30.51
15	19.49	19.53	19.57	19.61	19.65	19.69	19.73	19.77	19.82	19.85	19.89
20	14.30	14.34	14.36	14.39	14.42	14.45	14.48	14.50	14.53	14.55	14.58
25	11.20	11.22	11.24	11.27	11.29	11.31	11.33	11.34	11.36	11.38	11.40
30	9.14	9.16	9.17	9.19	9.20	9.22	9.23	9.24	9.26	9.27	9.29
35	7.67	7.68	7.69	7.70	7.72	7.73	7.74	7.75	7.76	7.77	7.78
40	6.57	6.58	6.59	6.60	6.61	6.62	6.62	6.63	6.64	6.65	6.66
45	5.72	5.73	5.74	5.74	5.75	5.76	5.76	5.77	5.78	5.78	5.79
50	5.05	5.05	5.06	5.06	5.07	5.07	5.08	5.08	5.09	5.09	5.09
55	4.50	4.50	4.51	4.51	4.51	4.52	4.52	4.52	4.53	4.53	4.53
60	4.04	4.05	4.05	4.05	4.05	4.06	4.06	4.06	4.06	4.06	4.07
65	3.66	3.66	3.67	3.67	3.67	3.67	3.67	3.67	3.67	3.67	3.67
70	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.34
75	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06	3.06

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	45	46	47	48	49	50	51	52	53	54	55
$(T_{CO} - T_{EV})$ °C	3.8871	3.9967	4.1085	4.2226	4.2226	4.4577	4.5788	4.7024	4.8283	4.9567	5.0876
10	1.34	1.33	1.33	1.33	1.32	1.32	1.32	1.32	1.31	1.31	1.31
15	1.56	1.55	1.55	1.54	1.54	1.53	1.53	1.52	1.52	1.51	1.51
20	1.83	1.82	1.81	1.80	1.79	1.79	1.78	1.77	1.76	1.76	1.75
25	2.15	2.14	2.13	2.12	2.11	2.09	2.08	2.07	2.06	2.05	2.04
30	2.56	2.54	2.52	2.50	2.49	2.47	2.45	2.44	2.42	2.40	2.39
35	3.06	3.03	3.01	2.98	2.96	2.93	2.91	2.89	2.86	2.84	2.82
40	3.69	3.65	3.62	3.58	3.54	3.51	3.48	3.44	3.41	3.38	3.35
45	4.48	4.43	4.38	4.33	3.28	4.23	4.18	4.14	4.09	4.05	4.01
50	5.49	5.41	5.34	5.27	5.20	5.14	5.07	5.01	4.95	4.89	4.83
55	6.78	6.68	6.58	6.48	6.39	6.29	6.20	6.11	6.03	5.95	5.86
60	8.45	8.30	8.17	8.03	7.90	7.77	7.65	7.53	7.41	7.29	7.18
65	10.65	10.44	10.24	10.06	9.86	9.69	9.51	9.35	9.19	9.03	8.87
70	13.54	13.24	12.98	12.70	12.46	12.21	11.96	11.72	11.50	11.27	11.06
75	17.40	17.01	16.60	16.24	15.86	15.53	15.17	14.86	14.52	14.23	13.94

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar)	65	66	67	68	69	70	71	72	73	74	75
$(T_{CO} - T_{EV})$ °C	6.5392	6.6993	6.862	7.030	7.197	7.370	7.543	7.722	7.901	8.086	8.271
10	31.11	31.13	31.20	31.27	31.31	31.36	31.38	31.44	31.47	31.52	31.56
15	20.24	20.26	20.30	20.34	20.37	20.40	20.41	20.45	20.46	20.50	20.52
20	14.82	14.84	14.86	14.88	14.90	14.92	14.93	14.96	14.97	14.99	15.01
25	11.57	11.58	11.60	11.61	11.63	11.64	11.65	11.67	11.67	11.69	11.70
30	9.41	9.42	9.43	9.44	9.45	9.46	9.47	9.47	9.48	9.49	9.50
35	7.87	7.88	7.89	7.89	7.90	7.91	7.91	7.92	7.92	7.93	7.93
40	6.72	6.73	6.73	6.74	6.74	6.75	6.75	6.75	6.76	6.76	6.76
45	5.83	5.83	5.84	5.84	5.84	5.85	5.85	5.85	5.85	5.85	5.86
50	5.12	5.13	5.13	5.13	5.13	5.13	5.13	5.13	5.13	5.13	5.13
55	4.55	4.55	4.55	4.55	4.55	4.55	4.55	4.55	4.55	4.55	4.55
60	4.07	4.07	4.07	4.07	4.07	4.07	4.07	4.07	4.07	4.06	4.06
65	3.67	3.67	3.67	3.67	3.67	3.67	3.66	3.66	3.66	3.66	3.65
70	3.33	3.33	3.33	3.33	3.33	3.32	3.32	3.32	3.32	3.31	3.31
75	3.04	3.04	3.04	3.04	3.03	3.03	3.03	3.02	3.02	3.02	3.02

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	65	66	67	68	69	70	71	72	73	74	75
$(T_{CO} - T_{EV})$ °C	6.5392	6.6993	6.862	7.030	7.197	7.370	7.543	7.722	7.901	8.086	8.271
10	1.29	1.28	1.28	1.28	1.28	1.28	1.27	1.27	1.27	1.27	1.26
15	1.47	1.46	1.46	1.46	1.45	1.45	1.44	1.44	1.44	1.43	1.43
20	1.68	1.68	1.67	1.66	1.66	1.65	1.65	1.64	1.64	1.63	1.63
25	1.94	1.93	1.92	1.91	1.90	1.90	1.89	1.88	1.87	1.86	1.86
30	2.25	2.32	2.22	2.21	2.20	2.19	2.17	2.16	2.15	2.14	2.13
35	2.62	2.60	2.58	2.57	2.55	2.53	2.52	2.50	2.48	2.47	2.45
40	3.07	3.05	3.02	3.00	2.97	2.95	2.93	2.91	2.88	2.86	2.84
45	3.62	3.59	3.56	3.52	3.49	3.46	3.43	3.40	3.37	3.34	3.31
50	4.30	4.26	4.21	4.17	4.13	4.08	4.04	4.00	3.96	3.92	3.88
55	5.15	5.09	5.02	4.97	4.91	4.85	4.79	4.74	4.69	4.63	2.58
60	6.21	6.12	6.04	5.96	5.88	5.80	5.73	5.65	5.58	5.51	5.44
65	7.54	7.42	7.31	7.20	7.10	6.99	6.89	6.79	6.70	6.61	6.51
70	9.23	9.08	8.92	8.78	8.63	8.49	8.36	8.23	8.10	7.97	7.85
75	11.40	11.19	10.99	10.79	10.59	10.40	10.22	10.04	9.87	9.70	9.53

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar)	55	56	57	58	59	60	61	62	63	64	65
$(T_{CO} - T_{EV})$ °C	5.0876	5.2210	5.3569	5.4954	5.6356	5.7802	5.9266	6.0757	6.2274	6.3820	6.5392
10	30.51	30.59	30.65	30.70	30.75	30.81	30.87	30.93	30.97	31.05	31.11
15	19.89	19.92	19.96	20.00	20.03	20.07	20.10	20.14	20.17	20.20	20.24
20	14.58	14.61	14.63	14.66	14.68	14.71	14.73	14.75	14.77	14.79	14.82
25	11.40	11.42	11.44	11.46	11.47	11.49	11.51	11.52	11.54	11.55	11.57
30	9.29	9.30	9.31	9.33	9.34	9.35	9.36	9.38	9.39	9.40	9.41
35	7.78	7.79	7.80	7.81	7.82	7.83	7.84	7.85	7.85	7.86	7.87
40	6.66	6.66	6.67	6.68	6.68	6.69	6.70	6.70	6.71	6.72	6.72
45	5.79	5.79	5.80	5.80	5.81	5.81	5.82	5.82	5.82	5.83	5.83
50	5.09	5.10	5.10	5.10	5.11	5.11	5.11	5.12	5.12	5.12	5.12
55	4.53	4.53	4.54	4.54	4.54	4.54	4.54	4.54	4.55	4.55	4.55
60	4.07	4.07	4.07	4.07	4.07	4.07	4.07	4.07	4.07	4.07	4.07
65	3.67	3.67	3.67	3.67	3.67	3.67	3.67	3.67	3.67	3.67	3.67
70	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.34	3.33
75	3.06	3.06	3.06	3.06	3.05	3.05	3.05	3.05	3.05	3.05	3.04

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	55	56	57	58	59	60	61	62	63	64	65
$(T_{CO} - T_{EV})$ °C	5.0876	5.2210	5.3569	5.4954	5.6356	5.7802	5.9266	6.0757	6.2274	6.3820	6.5392
10	1.31	1.31	1.30	1.30	1.30	1.30	1.29	1.29	1.29	1.29	1.29
15	1.51	1.50	1.50	1.50	1.49	1.49	1.48	1.48	1.47	1.47	1.47
20	1.75	1.74	1.73	1.73	1.72	1.71	1.71	1.70	1.69	1.69	1.68
25	2.04	2.03	2.02	2.01	2.00	1.99	1.98	1.97	1.96	1.95	1.94
30	2.39	2.37	2.36	2.34	2.33	2.32	2.30	2.29	2.27	.26	2.25
35	2.82	2.80	2.78	2.76	2.73	2.71	2.69	2.68	2.66	2.64	2.62
40	3.35	3.32	3.29	3.26	3.23	3.20	3.18	3.15	3.12	3.10	3.07
45	4.01	3.96	3.92	3.88	3.84	3.80	3.77	3.73	3.69	3.66	3.62
50	4.83	4.77	4.71	4.66	4.60	4.55	4.50	4.45	4.40	4.35	4.30
55	5.86	5.79	5.71	5.63	5.56	5.49	5.41	5.35	5.28	5.21	5.15
60	7.18	7.07	6.97	6.86	6.76	6.66	6.57	6.47	6.38	6.29	6.21
65	8.87	8.72	8.58	8.43	8.29	8.16	8.03	7.90	7.78	7.66	7.54
70	11.06	10.85	10.65	10.46	10.27	10.08	9.90	9.73	9.56	9.39	9.23
75	13.94	13.64	13.35	13.09	12.81	12.57	12.31	12.08	11.85	11.62	11.40

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar)	75	76	77	78	79	80	81	82	83	84	85
$(T_{CO} - T_{EV})$ °C	8.271	8.468	8.654	8.852	9.050	9.255	9.459	9.670	9.881	10.099	10.317
10	31.56	31.63	31.65	31.69	31.73	31.75	31.81	31.82	31.86	31.89	31.91
15	20.52	20.55	20.57	20.61	20.62	20.63	20.65	20.67	20.67	20.70	20.71
20	15.01	15.02	15.03	15.05	15.07	15.08	15.09	15.10	15.11	15.12	15.12
25	11.70	11.71	11.72	11.73	11.74	11.75	11.75	11.76	11.76	11.77	11.78
30	9.50	9.51	9.51	9.52	9.52	9.53	9.53	9.54	9.54	9.55	9.55
35	7.93	7.94	7.94	4.95	7.95	7.95	7.96	7.96	7.06	7.96	7.96
40	6.76	6.76	6.77	6.77	6.77	6.77	6.77	6.77	6.77	6.77	6.77
45	5.86	5.86	5.86	5.86	5.86	5.86	5.86	5.86	5.86	5.85	5.85
50	5.13	5.13	5.13	5.13	5.13	5.13	5.13	5.13	5.12	5.12	5.12
55	4.55	4.55	4.54	4.54	4.54	4.54	4.54	4.53	4.53	4.53	4.52
60	4.06	4.06	4.06	4.06	4.05	4.05	4.05	4.04	4.04	4.04	4.03
65	3.65	3.65	3.65	3.65	3.64	3.64	3.64	3.63	3.63	3.62	3.62
70	3.31	3.31	3.30	3.30	3.29	3.29	3.29	3.28	3.28	3.27	3.27
75	3.01	3.01	3.00	3.00	3.00	2.99	2.99	2.98	2.98	2.97	2.97

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar)	75	76	77	78	79	80	81	82	83	84	85
$(T_{CO} - T_{EV})$ °C	8.271	8.468	8.654	8.852	9.050	9.255	9.459	9.670	9.881	10.099	10.317
10	1.26	1.26	1.26	1.26	1.26	1.26	1.25	1.25	1.25	1.25	1.25
15	1.43	1.43	1.42	1.42	1.42	1.42	1.41	1.41	1.41	1.40	1.40
20	1.63	1.62	1.62	1.61	1.61	1.60	1.60	1.59	1.59	1.58	1.58
25	1.86	1.85	1.84	1.83	1.83	1.82	1.81	1.81	1.80	1.79	1.78
30	2.13	2.12	2.11	2.10	2.09	2.08	2.07	2.06	2.05	2.04	2.03
35	2.45	2.44	2.42	2.41	2.39	2.28	2.37	2.35	2.34	2.33	2.31
40	2.84	2.82	2.80	2.78	2.76	2.74	2.73	2.71	2.69	2.67	2.65
45	3.31	3.29	3.26	3.23	3.21	3.18	3.15	3.13	3.11	3.08	3.06
50	3.88	3.85	3.81	3.78	3.74	3.71	3.67	3.64	3.61	3.58	3.55
55	4.58	4.54	4.48	4.44	4.39	4.35	4.30	4.26	4.21	4.17	4.13
60	5.44	5.38	5.31	5.25	5.19	5.13	5.07	5.01	4.95	4.90	4.84
65	6.51	6.43	6.34	6.25	6.17	6.09	6.01	5.94	5.86	6.79	6.72
70	7.85	7.74	7.61	7.50	7.39	7.29	7.18	7.08	6.98	6.89	6.79
75	9.53	9.38	9.22	9.07	8.92	8.78	8.64	8.51	8.38	8.25	8.12

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar)	85	86	87	88	89	90	91	92	93	94	95
$(T_{CO} - T_{EV})$ °C	10.317	10.542	10.767	10.999	11.471	11.471	11.710	11.947	12.203	12.458	12.713
10	31.91	31.93	31.94	31.95	31.97	32.00	32.03	32.06	32.05	32.03	32.02
15	20.71	20.74	20.75	20.76	20.76	20.77	20.77	20.78	20.78	20.78	20.77
20	15.12	15.14	15.14	15.14	15.15	15.15	15.16	15.16	15.16	15.16	15.15
25	11.78	11.79	11.79	11.79	11.79	11.79	11.79	11.79	11.79	11.78	11.77
30	9.55	9.55	9.55	9.55	9.55	9.55	9.55	9.55	9.54	9.54	9.53
35	7.96	7.96	7.96	7.96	7.96	7.95	7.95	7.95	7.94	7.94	7.93
40	6.77	6.77	6.77	6.77	6.76	6.76	6.76	6.75	6.75	6.74	6.74
45	5.85	5.85	5.85	5.84	5.84	5.84	5.83	5.83	5.82	5.81	5.81
50	5.12	5.12	5.11	5.11	5.10	5.10	5.09	5.09	5.08	5.08	5.07
55	4.52	4.52	4.52	4.51	4.51	4.50	4.50	4.49	4.48	4.48	4.47
60	4.03	4.03	4.02	4.02	4.01	4.01	4.00	3.99	3.99	3.98	3.97
65	3.62	3.61	3.61	3.60	3.60	3.59	3.58	3.58	3.57	3.56	3.55
70	3.27	3.26	3.25	3.25	3.24	3.24	3.23	3.22	3.22	3.21	3.20
75	2.97	2.96	2.95	2.95	2.94	2.93	2.93	2.92	2.91	2.90	2.90

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C / (P_{CO} bar) / ($T_{CO} - T_{EV}$) °C	85	86	87	88	89	90	91	92	93	94	95
	10,317	10,542	10,767	10,999	11,471	11,471	11,710	11,947	12,203	12,458	12,713
10	1,25	1,24	1,24	1,24	1,24	1,24	1,24	1,24	1,23	1,23	1,23
15	1,40	1,40	1,39	1,39	1,39	1,39	1,38	1,38	1,38	1,38	1,37
20	1,58	1,57	1,57	1,56	1,56	1,56	1,55	1,55	1,54	1,54	1,54
25	1,78	1,78	1,77	1,77	1,76	1,75	1,75	1,74	1,74	1,73	1,72
30	2,03	2,02	2,01	2,00	1,99	1,98	1,98	1,97	1,96	1,95	1,94
35	2,31	2,30	2,29	2,28	2,27	2,25	2,24	2,23	2,22	2,21	2,20
40	2,65	2,64	2,62	2,60	2,59	2,57	2,56	2,54	2,53	2,51	2,50
45	3,06	3,04	3,01	2,99	2,97	2,95	2,93	2,91	2,89	2,87	2,85
50	3,55	3,52	3,49	3,46	3,43	3,40	3,37	3,35	3,52	3,30	3,27
55	4,13	4,09	4,05	4,02	3,98	3,94	3,91	3,87	3,84	3,80	3,77
60	4,84	4,79	4,74	4,69	4,64	4,59	4,55	4,50	4,46	4,41	4,37
65	5,72	5,65	5,58	5,51	5,45	5,39	5,32	5,26	5,20	5,15	5,09
70	6,79	6,70	6,61	6,52	6,44	6,36	6,27	6,20	6,12	6,04	5,97
75	8,12	8,00	7,88	7,77	7,66	7,55	7,44	7,34	7,24	7,14	7,04

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C	95	96	97	98	99	100	101	102	103	104	105
(P_{CO} bar)	12,713	12,976	13,238	13,509	13,779	14,058	14,337	14,625	14,912	15,209	15,505
($T_{CO} - T_{EV}$ °C)											
10	32,02	32,02	32,04	32,04	32,01	32,01	32,00	31,96	31,92	31,92	31,91
15	20,77	20,77	20,77	20,77	20,75	20,74	20,74	20,73	20,70	20,68	20,65
20	15,15	15,14	15,14	15,12	15,12	15,11	15,11	15,09	15,08	15,06	15,03
25	11,77	11,77	11,76	11,76	11,75	11,74	11,73	11,72	11,70	11,68	11,67
30	9,53	9,53	9,52	9,51	9,50	9,49	9,49	9,47	9,46	9,45	9,43
35	7,93	7,93	7,92	7,91	7,90	7,89	7,88	7,87	7,86	7,84	7,83
40	6,74	6,73	6,72	6,71	6,70	6,70	6,69	6,67	6,66	6,65	6,63
45	5,81	5,80	5,79	5,78	5,78	5,77	5,76	5,75	5,73	5,72	5,71
50	5,07	5,06	5,06	5,05	5,04	5,03	5,02	5,01	4,99	4,98	4,97
55	4,47	4,46	4,45	4,44	4,44	4,43	4,42	4,41	4,39	4,38	4,37
60	3,97	3,96	3,96	3,95	3,94	3,93	3,92	3,91	3,90	3,88	3,87
65	3,55	3,55	3,54	3,53	3,52	3,51	3,50	3,49	3,48	3,47	3,45
70	3,20	3,19	3,18	3,17	3,16	3,15	3,14	3,13	3,12	3,11	3,20
75	2,90	2,89	2,88	2,87	2,86	2,85	2,84	2,83	2,82	2,80	2,79

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar) / $(T_{CO} - T_{EV})$ °C	95	96	97	98	99	100	101	102	103	104	105
	12.713	12.976	13.238	13.509	13.779	14.058	14.337	14.625	14.912	15.209	15.505
10	1.23	1.23	1.23	1.23	1.23	1.23	1.22	1.22	1.22	1.22	1.22
15	1.37	1.37	1.27	1.37	1.36	1.36	1.36	1.36	1.36	1.35	1.35
20	1.54	1.53	1.53	1.53	1.52	1.52	1.52	1.51	1.51	1.51	1.50
25	1.72	1.72	1.71	1.71	1.70	1.70	1.69	1.69	1.68	1.68	1.68
30	1.94	1.94	1.93	1.92	1.91	1.91	1.90	1.89	1.89	1.88	1.87
35	2.20	2.19	2.18	2.17	2.16	2.15	2.14	2.13	2.12	2.11	2.10
40	2.50	2.49	2.47	2.46	2.44	2.43	2.42	2.41	2.39	2.28	2.37
45	2.85	2.83	2.82	2.80	2.78	2.76	2.75	2.73	2.71	2.70	2.68
50	3.27	3.25	3.22	3.20	3.18	3.15	3.13	3.11	3.09	3.07	3.05
55	3.77	3.74	3.71	3.68	3.65	3.62	3.59	3.56	3.53	3.51	3.48
60	4.37	4.33	4.29	4.25	4.21	4.17	4.13	4.09	4.06	4.02	3.99
65	5.09	5.04	4.98	4.93	4.88	4.83	4.78	4.73	4.69	4.64	4.60
70	5.97	5.90	5.83	5.76	5.69	5.63	5.57	5.51	5.44	5.39	5.33
75	7.04	6.95	6.86	6.77	6.68	6.60	6.52	6.44	6.36	6.29	6.21

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar)	105	106	107	108	109	110	111	112	113	114	115
$(T_{CO} - T_{EV})$ °C	15.505	15.811	16.116	16.431	16.746	17.071	17.395	17.730	18.064	18.409	18.753
10	31.91	31.91	31.86	31.81	31.76	31.71	31.59	31.54	31.46	31.39	31.31
15	20.65	20.64	20.59	20.59	20.55	20.52	20.45	20.41	20.35	20.30	20.23
20	15.03	15.02	15.00	14.97	14.94	14.91	14.87	14.83	14.80	14.75	14.71
25	11.67	11.66	11.63	11.61	11.59	11.56	11.53	11.50	11.46	11.43	11.39
30	9.43	9.41	9.40	9.37	9.35	9.33	9.30	9.28	9.25	9.22	9.18
35	7.83	7.82	7.80	7.78	7.76	7.74	7.72	7.69	7.66	7.64	7.61
40	6.63	6.62	6.61	6.59	6.57	6.55	6.53	6.51	6.49	6.46	6.43
45	5.71	5.70	5.68	5.66	5.65	5.63	5.61	5.59	5.57	5.54	5.52
50	4.97	4.96	4.94	4.93	4.91	4.89	4.88	4.86	4.84	4.81	4.79
55	4.37	4.36	4.34	4.33	4.31	4.30	4.28	4.26	4.24	4.22	4.20
60	3.87	3.86	3.84	3.83	3.81	3.80	3.78	3.76	3.75	3.73	3.71
65	3.45	3.44	3.43	3.41	3.40	3.38	3.37	3.35	3.33	3.31	3.29
70	3.10	3.09	3.07	3.06	3.04	3.03	3.01	3.00	2.98	2.96	2.94
75	2.79	2.78	2.77	2.75	2.74	2.72	2.71	2.69	2.68	2.66	2.64

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C (P_{CO} bar) / ($T_{CO} - T_{EV}$) °C	105	106	107	108	109	110	111	112	113	114	115
	15.505	15.811	16.116	16.431	16.746	17.071	17.395	17.730	18.064	18.409	18.753
10	1.22	1.22	1.22	1.22	1.22	1.21	1.21	1.21	1.21	1.21	1.21
15	1.35	1.35	1.35	1.35	1.34	1.34	1.34	1.34	1.34	1.34	1.33
20	1.50	1.50	1.50	1.49	1.49	1.49	1.49	1.48	1.48	1.48	1.48
25	1.68	1.67	1.67	1.66	1.66	1.65	1.65	1.65	1.64	1.64	1.63
30	1.87	1.87	1.86	1.86	1.85	1.84	1.84	1.83	1.83	1.82	1.82
35	2.10	2.10	2.09	2.08	2.07	2.06	2.05	2.05	2.04	2.03	2.03
40	2.37	2.36	2.35	2.34	2.33	2.32	2.31	2.30	2.29	2.28	2.27
45	2.68	2.67	2.65	2.64	2.62	2.61	2.60	2.58	2.57	2.56	2.54
50	3.05	3.03	3.01	2.99	2.97	2.95	2.94	2.92	2.90	2.88	2.87
55	3.48	3.45	3.43	3.40	3.38	3.36	3.33	3.31	3.29	3.27	3.24
60	3.99	3.96	3.92	3.89	3.86	3.83	3.80	3.77	3.74	3.71	3.69
65	4.60	4.56	4.51	4.47	4.43	4.39	4.35	4.32	4.28	4.24	4.21
70	5.33	5.27	5.22	5.16	5.11	5.06	5.01	4.96	4.92	4.87	4.82
75	6.21	6.14	6.07	6.00	5.93	3.87	5.80	5.74	5.68	5.62	5.56

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

T_{CO} °C (P_{CO} bar) / $(T_{CO} - T_{EV})$ °C	115	116	117	118	119	120	121	122	123	124	125
	18.753	19.109	19.464	19.831	20.197	20.575	21.952	21.342	22.731	22.133	22.535
10	31.31	31.16	31.05	30.92	30.79	30.65	30.55	30.38	30.22	3.01	29.93
15	20.23	20.16	20.08	20.01	19.92	19.84	19.72	19.63	19.51	19.44	19.31
20	14.71	14.66	14.60	14.54	14.48	14.41	14.33	14.26	14.18	14.12	14.03
25	11.39	11.35	11.30	11.26	11.20	11.16	11.10	11.04	10.97	10.92	10.85
30	9.18	9.15	9.11	9.07	9.03	8.98	8.93	8.88	8.83	8.78	8.73
35	7.61	7.58	6.55	7.51	7.47	7.43	7.39	7.35	7.30	7.26	7.22
40	6.43	6.41	6.38	6.34	6.31	6.28	6.24	6.20	6.16	6.12	6.08
45	5.52	5.50	5.47	5.44	5.41	5.38	5.34	5.31	5.27	5.24	5.20
50	4.79	4.77	4.74	4.72	4.69	4.66	4.63	4.60	4.56	4.53	4.50
55	4.20	4.18	4.15	4.13	4.10	4.07	4.05	4.02	3.98	3.96	3.92
60	3.71	3.69	3.66	3.64	3.62	3.59	3.56	3.54	3.51	3.48	3.45
65	3.29	3.27	3.25	3.23	3.21	3.18	3.16	3.13	3.10	3.08	3.05
70	2.94	2.92	2.90	2.88	2.86	2.84	2.81	2.79	2.76	2.74	2.71
75	2.64	2.62	2.60	2.58	2.56	2.54	2.52	2.49	2.47	2.44	2.42

THEORETICAL RANKINE COEFFICIENTS OF PERFORMANCE $(COP)_R$ FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114.

T_{CO} °C	115	116	117	118	119	120	121	122	123	124	125
(P_{CO} bar)	18.753	19.109	19.464	19.831	20.197	20.575	21.952	21.342	22.731	22.133	22.535
($T_{CO} - T_{EV}$) °C											
10	1.21	1.21	1.21	1.21	1.21	1.21	1.20	1.20	1.20	1.20	1.20
15	1.33	1.33	1.33	1.33	1.33	1.33	1.33	1.32	1.32	1.32	1.32
20	1.48	1.47	1.47	1.47	1.47	1.46	1.46	1.46	1.46	1.46	1.45
25	1.63	1.63	1.63	1.63	1.62	1.61	1.61	1.61	1.61	1.61	1.60
30	1.82	1.81	1.81	1.80	1.80	1.79	1.79	1.78	1.78	1.78	1.77
35	2.03	2.02	2.01	2.01	2.00	1.99	1.99	1.98	1.98	1.97	1.96
40	2.27	2.26	2.25	2.24	2.23	2.22	2.22	2.21	2.20	2.19	2.18
45	2.54	2.53	2.52	2.51	2.50	2.49	2.47	2.47	2.45	2.45	2.43
50	2.87	2.85	2.84	2.82	2.81	2.79	2.78	2.76	2.75	2.75	2.72
55	3.24	3.22	3.20	3.18	3.16	3.15	3.13	3.11	3.09	3.08	3.06
60	3.69	3.66	3.63	3.61	3.58	3.56	3.54	3.51	3.49	3.47	3.45
65	4.21	4.17	4.14	4.11	4.07	4.04	4.01	3.98	3.95	3.93	3.90
70	4.82	4.78	4.74	4.70	4.65	4.62	4.58	4.54	4.50	4.47	4.43
75	5.56	5.51	5.45	5.40	5.34	5.29	5.24	5.19	5.15	5.10	5.06

COMPRESSION RATIOS P_{CO}/P_{EV} FOR A RANGE OF LIFTS AND CONDENSING TEMPERATURES FOR R114

SYSTEM SPECIFICATION FOR MATCHED HEATING AND COOLING DUTY.

If the heat exchangers are of comparable cost per unit area the most cost effective method of design is to divide the available temperature drop into two equal parts. Thus, a table entry for a condensing temperature of (75 + X) degrees C. with a gross temperature lift of (37 + 2X) degrees C. which has a $(COP)_R$ at 3.60. Such a value occurs in the table, for Tco = 89 degrees C. and (Tco - Tev) = 65 degrees C. The system can, therefore, be approximately matched if the system is designed for approach temperature differences 14 degrees C. in each heat exchanger. From table this would require a compressor capable of a compression ratio of 5.45. It is necessary to recalculate the pressure drops based on the specific volumes of the vapours at these revised temperatures and to allow for the reduction in tube length permitted by the increased temperature drops. The $(HPE)_R$ value may then be recalculated and the procedure iterated to this point.

In designing the condenser it is safer to ignore the enhanced driving force provided by the super heat. If the heat transfer coefficients for boiling and condensation of R114 are available at 24 degrees C. and 89 degrees C respectively, the heat transfer areas required in the heat exchangers can be estimated by standard methods [8].

For safety, the condenser, evaporator and all instruments and piping should be capable of withstanding the maximum pressure which the duty requires of the system. Relief valves should be provided so that this pressure, here somewhat in excess of 11.5 bar, is not accidentally exceeded.

To proceed further with a design, table of thermo dynamic data, such as those used prepare the tables given in the chapter 5.1, are required.

From table it can be seen that second differences in the latent heat of vapourisation are approximately constant at temperatures near 15 degrees C. This implies that the data can be fitted by an equation quadratic in T degrees C. From the valves listed in table for 15 degrees C., 20 degrees C. and 25 degrees C. it is easy to derive a quadratic equation in T degrees C. for the latent heat as follows:

$132.76 = a\,(15)^2 + b\,(15) + C$ KJ kg-1

$130.853 = a\,(20)^2 + b\,(20) + C$ KJ kg-1

$128.842 = a\,(25)^2 + b\,(25) + C$ KJ kg-1

$225a + 15b + C = 132.76$

$400a + 20B + C = 130.853$

$625a + 25 + C = 128.842$

$$D = \begin{vmatrix} 225 & 15 & 1 \\ 400 & 20 & 1 \\ 625 & 25 & 1 \end{vmatrix} = -250$$

$$D_a = \begin{vmatrix} 132.76 & 15 & 1 \\ 130.853 & 20 & 1 \\ 128.842 & 25 & 1 \end{vmatrix} = 0.52$$

$$D_b = \begin{vmatrix} 225 & 15 & 132.76 \\ 400 & 20 & 130.853 \\ 625 & 25 & 128.842 \end{vmatrix} = 77.275$$

$$D_c = \begin{vmatrix} 225 & 15 & 132.76 \\ 400 & 20 & 130.853 \\ 625 & 25 & 128.842 \end{vmatrix} = -34464.23$$

$$a = \frac{D_2}{D} = \frac{-0.52}{250} = -0.00208$$

$$V = \frac{D_b}{D} = -\frac{77.275}{250} = -0.3091$$

$$V = \frac{D_c}{D} = -\frac{34464.23}{250} = 137.856\ 92$$

It follows that latent heat of vapourisation near 15 degrees C.

$$-0.00208\ T^2 - 0.3091T + 137.85692\ \ KJ\ \ kg\text{-}1$$

In this way the following relationships could be derived from the data of the table.

(a) latent heat of vapourisation near 89 degrees C.

$a_2\ (T-89)^2 + b_1\ (T-89) + c$ KJ kg-1

(b) enthalpy of saturated vapour near 15 degrees C.

$a_2\ T^2 + b_2\ T + c_2$ KJ kg-1

(c) enthalpy of saturated vapour near 89 degrees C.

$a_3\ (T-89)^2 + b^3\ \ T + c_3$ KJ kg-1

(d) vapor density near 15 degrees C.

$a_4\ T^2 + b_4\ T + c_4$ kg m^{-3}

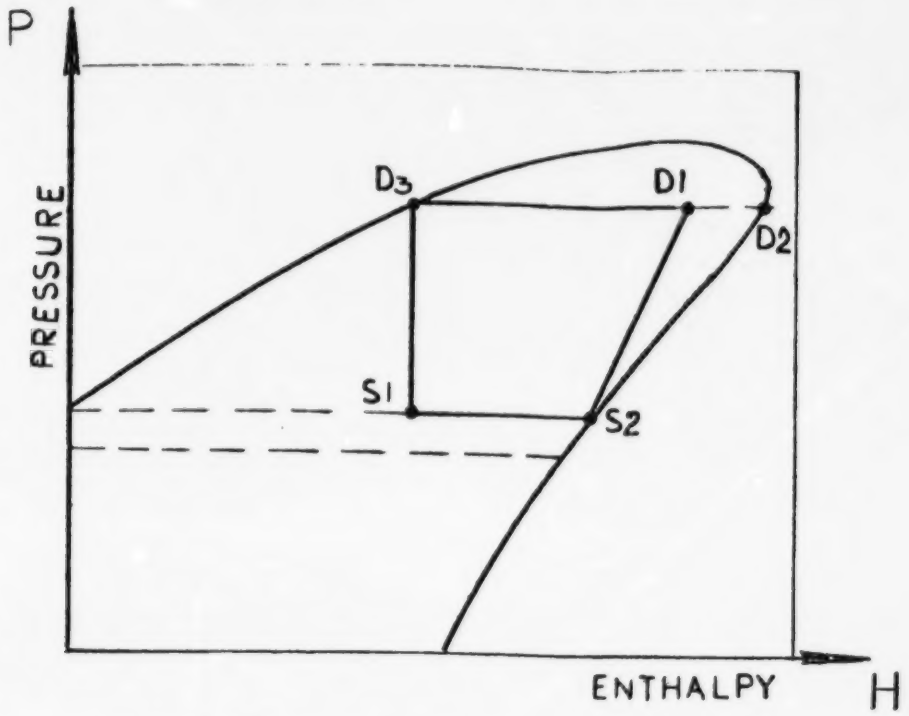
(e) vapor density near 89 degrees C.

$a_5\ (T-89)^2 + b_5\ \ (T-89) + c_5$

These equations may be used to estimate the date of 15 degrees C. and of 89 degrees C.

If saturated liquid were expanded isenthalfically through the expansion valve the wetness fraction X of the vapour leaving the expansion valve can be estimated from equation

$$H_{D1} = H_{D3} \times X + H_{D2} \times (1-X)$$



enthalpy per unit mass H, kJ kg^{-1}

Over the expansion range 89 degrees C. to 15 degrees C. the wetness fraction becomes

$$X = \frac{H_{D1} - H_{D2}}{H_{D3} - H_{D2}}$$

Enthalpy of saturated vapour at 24 degrees C = 254.084 KJ kg-1

Enthalpy of saturated liquid at 89 degrees C. 129.806 K kg-1

The energy to be supplied in the evaporator is:

124.278 Ky Kg-1

Based on this value the required circulation rate of working fluid is:

$Q_S/124.278 = 2.4$ Kg S^{-1}

This corresponds to a volumetric flow rate of 0.1517 M^3 S^{-1} at the compressor inlet.

The compressor has to be capable of moving these volumes with a compression ratio of 5.45. The above vapour velocities may be used to estimate the pressure drops in the pipe work and in the heat exchangers.

In practice the vapour at the compressor inlet is usually super heated by between 5 degrees C. and 25 degrees C.

For a real gas, the gas low is modified to
PV (MW) = ZRT (5.1)

When Z is the compressibility factor R is the universal gas constant = 8.315 KJ Kmd^{-1} K^{-1} and T is the absolute temperature.

For R114 the inolecular weight (MW) = 170.9 Ky Kmd^{-1} so that the specific gas constant per unit mass

$R^1 = 8.315 / 170.9 = 0.0486$ K Kg^{-1} K^{-1}

From the table PV = 0.1345 bar m^3 K^{-1} at 24 degrees C. = 297 degrees K thence, the compressibility factor at 0 degrees C. is:

$Z = 13.45/0.0486 \times 297 = 0.9318$

Similarly at 35 degrees C. Z = 0.9027 and 40 degrees C. Z = 0.915. These value vary linearly with temperature. Super heating the vapour from 24 degrees C. to 45 degrees C. will, therefore, increase the volumetric vapor rate at the compressor inlet.

By using the standard thermo dynamic relationships other information can be estimated from the tables. The polytropic coefficient of expansion of a real gas.

K may be defined by equation

$PV^k = \text{const}$ (5.2)

If the compressibility factor can be assumed constant it may be deduced that the isentropic work of compression Wi between states 1 and 2 is given by equation.

$$Wi = \frac{RT2}{n}\left(\frac{P1}{P2}\right)^n - 1 = \frac{RT2}{n}\left(\frac{T1}{T2}\right)_{-1} \quad (5.3)$$

Where n = (K-1)/K and the temperatures and pressures are in absolute units.

From the table at T = 15 degrees C. $_1$ P = 1.5193 bar and at T = 20 degrees C_1P = 1.8047

From equations K5.1) and $(5.2)_1$ at constant z

$(T2/t1) = (P2/P1)^n$

Substitute values to give

$$n = \frac{(K-1)}{K} = \frac{[\ln (20 + 273.16) / (15 + 273.16]}{\ln (1.8047 / 1.5193)} = 0.1$$

From which $K = (1-0.1)^{-1} = 1.111$ in the range 15 degrees to 20 degrees C. In the same way, over compression range 24 degrees C. to 89 degrees C $K = 1.125$

Value of Wi may be iterated if required.

The least reliable estimate from the data presented in the tables is that of the heat capacity per unit mass of vapor at constant pressure. The best estimate possible is given by equation.

$c_p = R1/ (Z \times n)$.

5.2 Effect of Subcooling and Super Heating of Refrigerant in the Heat Pump Cycle

The pressure-Entholphy diagram for R114 shows that the saturated liquid line has a relatively small gradient. It implies that a partial condensation will result in an isentropic compression of the saturated vapor. However, in practice the vapor at the suction of compressor is usually superheated by about 5 degrees C. to 25 degrees C. to avoid the possibility of the condensation of entrained liquid droplets in the compressor.

In a super-heated and sub-cooled cycle, from point (1) to point (1'), the saturated vapor is super-heated. The refrigerant is compressed from point (1') to point (2) through the compressor. From point (2) to point (3) condensation takes place in the two condensers and heat is rejected to evaporate moisture. From point (3) to point (3') the liquid refrigerant is further cooled down by the sub-cooler. From point (3') to point (4) the refrigerant expands through the thermo-expansion valve. The refrigerant carries only a very small quantity of vapor at this stage.

From point (4) to point (1) evaporation takes place and heat is absorbed.

In our system, the vapor at the suction must be superheated by a minimum of 18 degrees C.

The improvement in the theoretic capacity of the thermal cycle can be determined as follows:

Q_s = amount of heat extracted
1019288.6 Btu/hr.

T_1 = suction temperature of compressor
45 degrees c.

T_2 = discharge temperature of compressor
89 degrees C.

M = mass flow rate of the refrigerant

Flow rate of refrigerant is

$$M = \frac{Q_s}{h-h4}$$

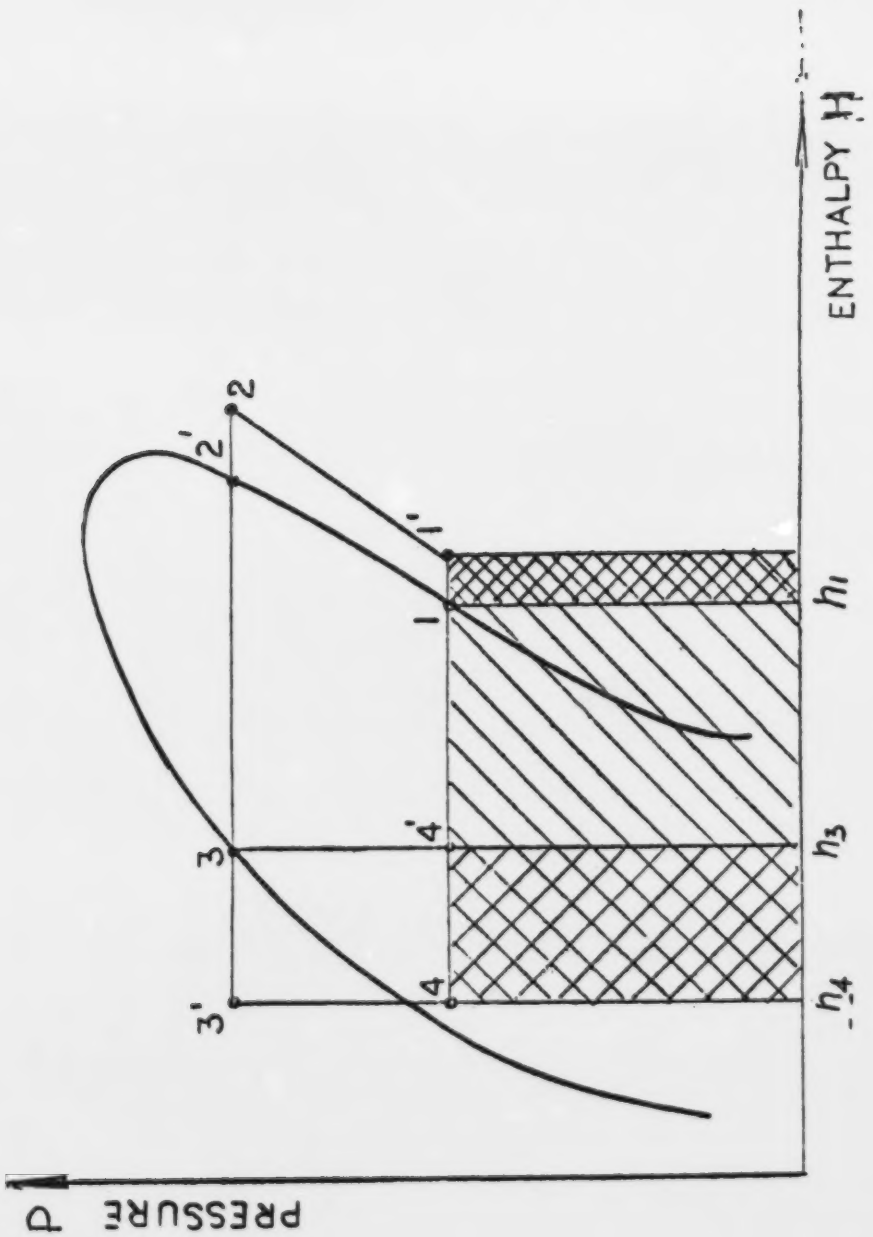
$$= \frac{1019288.6}{84-34}$$

= 20384.5 lb./hr.
= 9265.7 Kg/hr.

heat extracted without sub-cooling

$$Q_s = M\ (h1 - h3)$$
$$= 20384.5\ (84-61)$$
$$= 468843.5 \text{ Btu/hr.}$$

$$Q_{AD} = M \times (h3 - h3')$$
$$= 20384.5\ (56-34)$$
$$= 448459 \text{ Btu/hr.}$$


PRESSURE P
ENTHALPY H
1
1'
2
2'
3
3'
4
4'
h1
h3
h4

Increase of overall efficiency is determined as:

$$= \frac{1019228.6 - 448959}{1019228.6}$$

$$= 50\%$$

5.3 Required Circulation Rate of Air Over the Evaporation For a Defined Dehunidification Load

In accordance with the data provided by Canada Sandpaper 470 lb./hr. of moisture is to be removed. On the Enthalpy-Humidity Psychrometric Chart, we can show that the dehumidification process takes place from point 2 to point 1 by condensation of the moisture on the evaporator coils. At point 2, the condition of air before the evaporator is 25% relative humidity. 166 degrees F. temperature, 0.0625 lb. of moisture per lb. of dry air.

At point 1, the condition of air after removal of moisture is 100% relative humidity, 80 degrees F. temperature, 0.0225 lb. of moisture per lb. of dry air.

Reduction of moisture content from point 2 to point 1

= 0.0625 - 0.0225 lb./lb. of air

= 0.04 lb./lb. of air

Required moisture removal rate 470 lb./hr.

Flow rate of air required

$$= \frac{470}{0.04}$$

= 11750 lb./hr.

= 195.8 lb./min.

= 4000 CFM

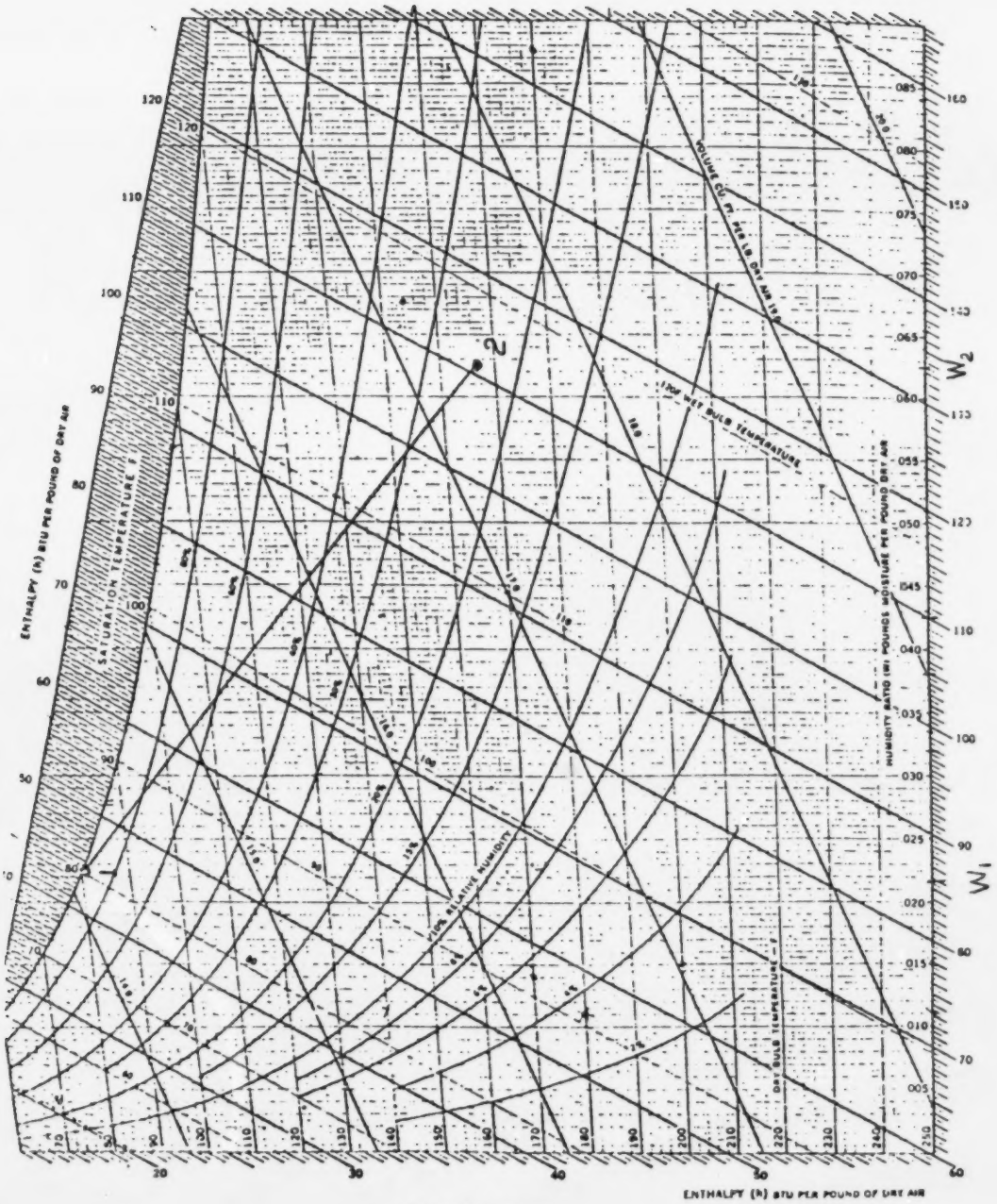

ENTHALPY (h) BTU PER POUND OF DRY AIR
SATURATION TEMPERATURE F
HUMIDITY RATIO (W) POUNDS MOISTURE PER POUND DRY AIR
DRY BULB TEMPERATURE — F
ENTHALPY (h) BTU PER POUND OF DRY AIR
W1
W2

FIGURE E-9

6.0 HEAT PUMP DEHUMIDIFICATION SYSTEM.

6.1 Drying by Conduction and Forced Convention

A high temperature heat pump can be installed into the existing Canada Sandpaper facility in Plattsville, Ontario with only minor modification to the present system. The driving force of this heat pump dehumidification system is provided by two 125 H.P. reciprocating compressors (item 1). On the discharge side of compressors, two different condensers are being connected in parallel.

The first one is a plate coil condenser (item 3) which is designed and built to fit into the existing conveyor system. The width of this condenser is determined by the smallest width of the backing cloth in production. The flat surface of the condenser provides excellent contact area for heat transfer to the wet cloth which is being stretched and fed over this condenser. Evaporation of moisture takes place on the top surface of the condenser while condensation of refrigerant occurs on the inside surface. Since these two processes which involve latent heat of evaporization, occur simultaneously, excellent heat transfer efficiency is achieved. To prevent the formation of scale or rust on its surface, this plate coil condenser must be made of stainless steel material.

The second one is the air cooled condenser (item 2) which is being mounted between the two ductworks to zone 1 and zone 2. This condenser supplies hot air through the nozzle jets on to the wet cloth.

The reason for the requirement of this additional condenser is as follows. In some cases of operation the width of wet cloth could be larger than that of the plate coil condenser. It

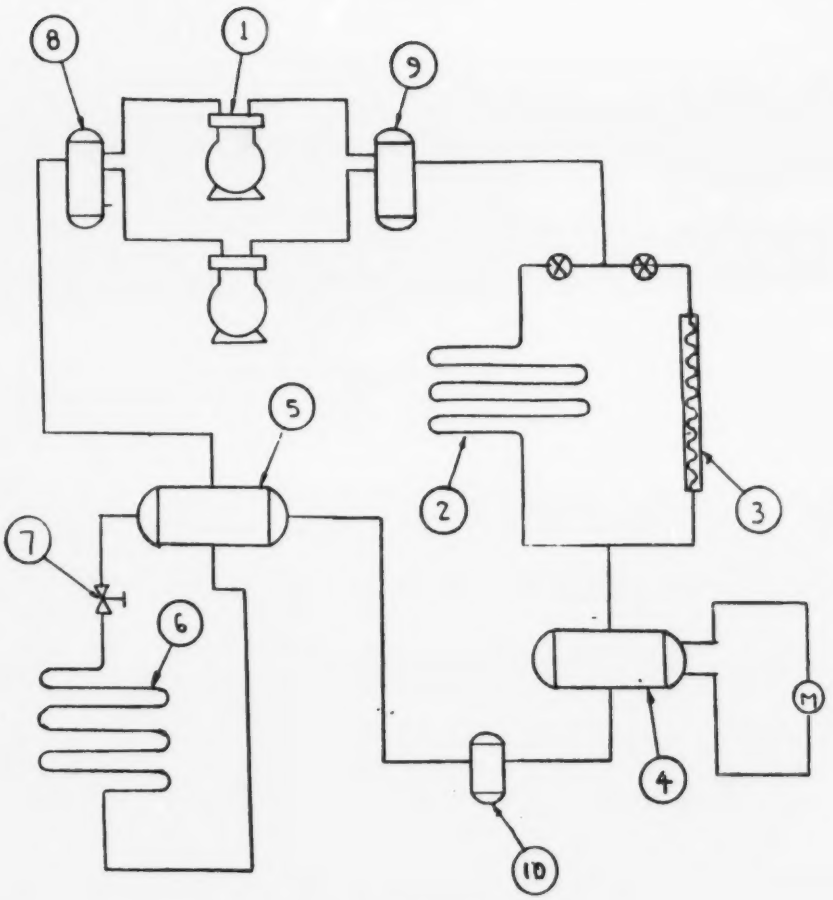
means that the edges of the wet cloth cannot be dried by direct conduction of heat.

To compensate for this drawback, the air cooled condenser is required to supply hot air for drying by forced convection. These two condensers are being operated at a temperature of 190 degrees F.

6.2 Sub-Cooler and Suction-Line Heat Exchanger

In order to improve the thermal efficiency of the overall system, a sub-cooler (item 4) is installed downstream of the condensers. After condensation, the liquid refrigerant is cooled so that its temperature is reduced below the saturation temperature. This sub-cooler is a shell and tube type heat exchanger, using the make-up water for the steam or hot water boilers as secondary refrigerant. The heat removed from the hot refrigerant can be used to reduce the energy cost of the plant.

In order to prevent liquid refrigerant droplets from condensing in the cylinders of the compressor, a suction-line heat exchanger (item 5) is installed upstream of the evaporator. This super-heater is a shell and tube heat exchanger. The liquid refrigerant leaving the sub-cooler (item 4) is used to super-heat the vapor refrigerant before it enters into the suction of compressors. The heat exchanger is to raise the temperature of refrigerant vapor above its saturation temperature. For the super-heated cycle, a greater quantity of heat must be rejected at the condensers per pound than for the saturated cycle.



PART NAME

1. COMPRESSOR
2. AIR-COOLED CONDENSER
3. PLATE-COIL CONDENSER
4. SUB-COOLER
5. SUCTION-LINE HEAT EXCHANGER
6. FIN-COIL EVAPORATOR (DEHUMIDIFIER COIL)
7. THERMISTIC EXPANSION VALVE
8. LIQUID SEPARATOR
9. OIL SEPARATOR
10. SEPARATOR

6.3 Dehumifification by Fin-Coil Evaporator

To prevent the humidity from building up in the oven, the moisture evaporated from the wet cloth must be removed from the air.

The cool surface of the fin-coil evaporator (item 6) provides excellent contact area for the moisture to condense. The evaporator, being mounted between the ductworks from the two drying zones will have portion of air diverted to it by a damper.

As the humid air is continuously being circulated over the fin-coils, the moisture is condensed on the cool surfaces of the coils and drained accordingly. To achieve high efficiency of dehumidification, a thermo expansion valve is installed upstream of the evaporator which is being operated at temperature of 80 degrees F.

In this close loop dehumidification system, the moisture is continuously removed from the air stream by condensation and simultaneously, the heat of evaporization is provided by the two condensers. Both the latent and sensible heat for drying are conserved in the oven. The result is a substantial saving in the cost of energy as the need for excessive exhaust is eliminated. The only energy input to the system is that required by the compressors, the blower, and the fan of the air-cooled condenser. Moreover the heat energy extracted from the sub-cooler can be recovered by the steam boilers and other facility of the plant.

Another aspect to be considered is how to maintain the relative humidity of the air in the oven. Since the rate of evaporation is a function of feed rate of the wet cloth,

decrease of the feed speed of the conveyor system will reduce the relative humidity of the air.

By adjustment of the feed speed of the conveyor, the level of relative humidity can be maintained.

6.4 Design and Construction of the Heat Pump Dehumifification System

The basic components of the heat pump dehumidification system are as follows:

(1) Reciprocating Compressors

2 Carrier Model # 5H126 - 125 H.P.
Compressors for R114 refrigerant
All compressors shall be complete with suction and discharge stop valves
Suction strainer
Forced feed lubrication system
High to low integral relief valve
Differential oil failure protection switch
Oil pressure gauge
One step manual capacity reduction
Removal cylinder liners
Compressor jacket water solenoid valve
V-belt drive and belt guard
Motor, pulley, and flywheel
High and low pressure cutouts
Hand oil pump for changing oil
Special tools.

(2) Air-cooled Condenser.

Dunham-Bush Model # L S BC 221B.

Capacity: 735000 Btu/hr. @ 10 degrees C.

Size: 86-3/4" x 130-1/2" x 61"

This unit shall be complete with two 2 H.P. fans with blades constructed of heavy duty aluminum.

All fans shall be statically balanced and be protected with heavy gauge steel guards.

All coils are constructed of copper tubing mechanically bonded to aluminum fins.

(3) Plate - Coil Condenser.

This heat exchanger is of stainless steel construction with flat surface on one side and corrugated surface on the other side. The width of the condenser is determined by the smallest size of cloth in production and the length is about 45'.

The flat surface on top provides full contact heat transfer between the refrigerant R114 and the wet cloth. Design pressure and design temperature is 350 PSIG and 250 degrees F. respectively. Tranter custom design model.

(4) Sub-Cooler.

Dunham-Bush Model # CSTC - 1048.

Size: 10-3/4" O.D. x 58" LG.

This shell and tube type heat exchanger is designed and built to ASME pressure vessel code.

The vessel is of carbon steel construction and is horizontally mounted.

It shall be complete with 1-1/4" inlet and outlet connections for water, and 1-3/8" inlet and 1-1/8" outlet connections for refrigerant.

Pressure drop at 37 GPM is approximately 3.5 PSI.

(5) Suction-Line Heat Exchanger.

Dunham Bush Model # 140 Rx.

Size: 12-1/4" O.D. x 29-1/2 LG.

This shell and tube type heat exchanger is designed and built to ASME Pressure Vessel Codes.

The vessel is of carbon steel construction and is horizontally mounted.

It shall be complete with 5" inlet and outlet connections for the shell side and 2" inlet and outlet connections from the tube side.

(6) Fin-Coil Evaporator.

Dunham-Bush model RU 3000-8

Size: 139" x 79" x 6"

The unit shall be complete with two 3 H.P. fans.

All fans shall be statically and dynamically balanced for quiet operation.

The coils are constructed of seamless copper tubings mechanically bonded to aluminum fins.

7.0 COMPARISON OF HIGH TEMPERATURE HEAT PUMP DEHUMIDIFIER WITH EXISTING STEAM-HEATED CONVECTION DRYING SYSTEM.

Steam-Heated Dryer Gas-Fired Boiler	GAS USAGE = 9257883 Btu/hr. = 9.768 x 10^6 KJ/Hr.
	Electrical power consumption = 49.22 KW = 1.772 x 10 KJ/Hr.
	Water removal = 213 KG/hr.
	Specific energy consumption = 46691 KJ/KG.
Heat Pump Dehumidifier System Unit Capacity = 245 KW	
Capital cost $250,000	Total Power Consumption = 453 KW = 1.631 x 10^6 KJ/Hr.
	Water removal = 213 KG/Hr.
	Specific energy consumption = 7656 KJ/KG.
Comparison	Energy usage reduction 83.6%
	Energy cost reduction = 53%
	Savings per year $82,250.
	Payback period 3.04 years At 1982 fuel price level

Notes:

The saving in the energy costs of $82,250 a year is based on the condition that the system is being operated 16 hours a day, five days a week, and 50 weeks a year. The total operating time is 4000 hours per year.

The condition used for comparison is based on winter period with an average ambient temperature of 15 degrees F. The payback period for a capital investment of $250,000 to install the additional heat pump dehumidifier system is estimated as approximately 3 years based on the above said conditions.

ENERGY COSTS BASED ON WINTER CONDITION:

Steam-heated System

Electrical power consumption for (4) 17000 CFM fans and (2) 10300 CFM fans.

= 66 H.P.

= 49.22 KW.

Heat loss = 711303 Btu/Hr.

Energy for pre-heating and evaporation = 526500 Btu/Hr.

Energy required by make-up air = 8020080 Btu/Hr.

Total heat input = 711303 + 526500 + 8020080

= 9257883 Btu/Hr.

Dehumidifier System

Electrical power consumption for (2) 125 h.p. compressors, (2) 2 h.p. fans (2) 3 h.p. fans for the evaporator = 196.12 KW.

Electrical power consumption for fans, blowers = 49.22 KW.

Total Electrical power consumption (196.12 + 49.22) KW.
= 245.34 KW.

In Ontario cost of electricity is 6.2 cents per KWH.

Total energy cost = $72,382 per year.

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